PB-244 306

PORTABLE AIR COMPRESSOR NOISE: DIAGNOSIS AND CONTROL

W. N. Patterson, et al

Bolt Beranek and Newman, Incorporated

Prepared for:

Environmental Protection Agency

29 March 1974

DISTRIBUTED BY:

NTIS

National Technical Information Service
U. S. DEPARTMENT OF COMMERCE

237145

BOLT BERANEK AND NEWMAN INC

CONSULTING • DEVELOPMENT • RESEARCH

PB244306

Report No. 2795b

PORTABLE AIR COMPRESSOR NOISE: DIAGNOSIS AND CONTROL

W.N. Patterson
B.S. Murray
F.R. Kern, Jr.
M.N. Alakel
J.I. Smullin
H.L. Fox

29 March 1974

Submitted to:

Environmental Protection Agency
Room 1122, Building 2
Crystal Mall
1921 Jefferson Davis Highway
Arlington, Virginia 20460

Attn: Eugene Burcher

Reproduced by
NATIONAL TECHNICAL
INFORMATION SERVICE
U S Department of Commerce
Springfield VA 22151

142

BIBLIOGRAPHIC DATA SHEET

1. Report No. 2795b

2.

PB 244 306

3. Report Date 29 March 1974

4. Title and Subtitle

Portable Air Compressor Noise: Diagnosis and Control

6.

7. Author(s) W. N. Patterson, B. S. Murray, F. R. Kern, Jr., M. N. Alakel, J. I. Smullin, H. L. Fox

8. Performing Organization Rept. No. 2795b

9. Performing Organization Name and Address

Bolt Beranek and Newman Inc.
50 Moulton Street
Cambridge, Massachusetts 02138

10. Project/Task/Work Unit No. 68-01-2316(002)

11. Contract/Grant No. EPA 68-01-1547

12. Sponsoring Organization Name and Address

Environmental Protection Agency
Office of Noise Abatement and Control
Washington, D.C. 20460

13. Type of Report & Period Covered Final Jan.-March 1974

14.

15. Supplementary Notes

16. Abstracts

In this report, the results of diagnostic evaluations of the noise emissions of three quieted portable air compressors are presented. The report presents an assessment of the cost to manufacturers to quiet the compressors tested. This assessment incorporates the following: (1) material and direct labor costs to manufacturers; (2) a noise source diagnosis and an operational performance diagnosis for each unit tested; (3) the estimated quieting costs for alternative designs; and (4) an assessment of the available foreign technology and associated costs to quiet portable air compressors.

17. Key Words and Document Analysis. 17a. Descriptors

- gasoline and diesel engine-powered air compressor
- noise aspects of design
- testing and results
- individual noise sources
- noise control techniques and reduction programs
- ranking of noise sources
- operational diagnosis
- noise characteristics
- evaluation of present quieting
- foreign air compressors
- cost of quieting

17b. Identifiers/Open-Ended Terms

17c. COSATI Field/Group

PRICES SUBJECT TO CHANGE

18. Availability Statement Release unlimited

19. Security Class (This Report) UNCLASSIFIED

20. Security Class (This Page) UNCLASSIFIED

21. No. of Pages 142

22. Price $5.75 2.25

FORM NTIS-35 (REV. 3-72)

THIS FORM MAY BE REPRODUCED

USCOMM-DC 14952-P72

PREFACE

In Report No. 2795b, we assess the technology and costs of quieting portable air compressors under the auspices of the Environmental Protection Agency [Contract 68-01-1547, Task 68-01-2316 (002)]. Research was conducted in January, February, and March, 1974.

We should like to thank two air compressor manufacturers for assistance in this project. For testing, Worthington CEI Inc. provided, free of charge, two air compressors: a 160 QT and a 750-QTEX; Atlas Copco Inc. rented a VSS-170 Dd unit. In addition, both companies supplied information on the technology and costs of their products.

Preceding page blank

TABLE OF CONTENTS

page

PREFACE iii

LIST OF FIGURES ix

LIST OF TABLES xiii

SECTION 1. INTRODUCTION 1

2. GASOLINE ENGINE-POWERED AIR COMPRESSOR 3
  2.1 The Worthington 160 QT Air Compressor 3
  2.2 Noise Aspects of the Design 3
  2.3 Testing 7
    2.3.1 Test program 7
  2.4 Test Results 8
  2.5 Individual Noise Sources 18
    2.5.1 Engine and compressor case-radiated noise 18
    2.5.2 Engine cooling fan 18
    2.5.3 Exhaust and muffler noise 25
    2.5.4 Compressor and engine air intake ... 27
    2.5.5 Side-curtain effect 30
  2.6 Noise Control Techniques 30
    2.6.1 Effectiveness of noise control techniques 30
    2.6.2 Operational diagnosis 32
  2.7 Further Noise Reduction Programs 32

3. DIESEL ENGINE-POWERED AIR COMPRESSOR: 500 cfm OR LESS 35
  3.1 The Atlas Copco VSS-170 Dd Compressor 35
  3.2 Testing 35
  3.3 Ranking of Noise Sources 38
  3.4 Noise Control Techniques 44

TABLE OF CONTENTS *(Cont.)*

page

3.5 Possible Additional Noise Control Techniques 52

3.6 Operational Diagnosis 53

3.6.1 Cooling system performance 53

3.6.2 Fuel consumption 53

3.6.3 Physical integrity of the enclosure 53

3.6.4 Effects of additional noise control on fuel consumption and cooling 54

SECTION 4. DIESEL ENGINE-POWERED AIR COMPRESSOR: >500 cfm 55

4.1 Introduction 55

4.2 Present Noise Characteristics 58

4.3 Identifying the Relative Contribution of the Major Sources of Noise 65

4.3.1 Fan noise 67

4.3.2 Other noise sources 69

4.4 Evaluation of Present Quieting 75

4.4.1 Acoustic enclosure 75

4.5 Determination of a Strategy for Further Quieting 78

4.5.1 Enclosure redesign 78

4.5.2 Performance of an improved fan design 80

4.5.3 Exhaust-related noise reduction 83

4.6 Quieting to 65 dBA 84

5. FOREIGN AIR COMPRESSORS 85

5.1 European Manufacturers 85

5.2 Noise Control Technology 88

5.3 The Cost of Quieting Foreign Air Compressors 89

5.4 Competition by Foreign Air Compressor Manufacturers 91

TABLE OF CONTENTS *(Cont.)*

page

SECTION 6. CONCLUSIONS 95

REFERENCES 97

APPENDIX A: COST OF QUIETING 99

B: PROJECTED COSTS TO REDUCE COMPRESSOR NOISE TO 65 dBA 109

C: COMPARISON OF OPERATIONAL PERFORMANCE 117

GLOSSARY OF TECHNICAL TERMS G-1

LIST OF FIGURES

page

Figure 1. General view of Worthington 160 QT 4

2. Worthington 160 QT with side curtain opened 4

3. Closeup of Worthington 160 QT engine and compressor compartment 5

4. Worthington 160 QT main design features 6

5. Compressor installation in semianechoic room ... 9

6. Total wrapping of unit for initial source diagnosis 9

7. Muffler shell uncovered 10

8. Fan drive motor and blanking panel 10

9. Individual source contributions in 1/3-octave band and overall levels, measured 7 m from right side of Worthington 160 QT compressor 12

10. Summary of source contributions measured at 7 m from right side of compressor 14

11. Typical variation in 1/3-octave band spectra for two measurement locations on 7-m radius hemisphere. For location of positions, see table in Report No. 2795a, Table 3 15

12. Constant bandwidth analysis (4 Hz) of sound measured 6.4 m from left side of fan 16

13. Pure tone directivity - fan, engine, and compressor 17

14. One-third octave band and overall fan noise levels measured 7 m from right side of compressor 19

15. One-third octave band and overall fan noise levels measured 7 m from rear of compressor 21

LIST OF FIGURES *(Cont.)*

page

Figure 16. Engine cooling fan and shroud 23

17. Cooling fan operating conditions 24

18. Firing frequency noise variation with distance and height above surface (A-weighted) 26

19. Compressor and engine intake silencer 28

20. One-third octave band and overall engine and compressor intake noise levels measured 7 m from right side of compressor 29

21. Effect of side curtain door on 1/3-octave band and overall noise levels measured 7 m from right side of compressor 31

22. Atlas-Copco air compressors: on top, super silensair; below, standard model 36

23. Anechoic chamber with compressor installed showing measurement positions 37

24. Four views of test configuration used in semi-anechoic chamber 39

25. Spectral measurements of unenclosed unit at 1 m 41

26. Measured sound level and A-weighted spectrum at 7 m from right side 43

27. Contributions to dBA ISO-2151-1972 sound level at 7-m distance 45

28. Exhaust silencer 46

29. Views of acoustic enclosure: side walls and end doors 47

30. Engine intake and cooling air ducts 48

31. Compressor intake and cooling air ducting 49

LIST OF FIGURES *(Cont.)*

page

Figure 32. Engine and compressor cooling air exhaust ducts 50

33. Noise reduction 7 m from right side when access doors are closed 51

34. Worthington "Blue Brute" model 750-QTEX, a quieted diesel engine powered compressor delivering 750 cfm 56

35. Quieting mechanisms used on Worthington model 750-QTEX 57

36. Summary data on noise emissions of Worthington "Blue Brute" model 750-QTEX 59

37. Linear and A-weighted 1/3-octave band noise spectra at four cardinal points and overhead at a distance of 7 m from the Worthington "Blue Brute" model 750-QTEX 61

38. Octave band spectra at three points on a 10-meter hemisphere showing general uniformity of radiation from Worthington "Blue Brute" model 750-QTEX 62

39. A-weighted octave band spectra at three heights showing typical variation with height at 7 m from side of Worthington "Blue Brute" model 750-QTEX 63

40. A-weighted 1/3-octave band spectra of the sound pressure level at 4, 8, and 16 m along a 270° radius from the center of the Worthington 750-QTEX 64

41. A-weighted 1/3-octave band and overall noise levels from the Worthington 750-QTEX for three operating conditions: idle and full rated delivery with doors closed, and full rated delivery with doors open 66

42. Narrowband spectra of two sources of the noise emission of the Worthington 750-QTEX 68

LIST OF FIGURES *(Cont.)*

page

Figure 43. Comparison of two positions of A-weighted octave band spectra of full power noise and noise generated by the fan alone 70

44. A-weighted overall levels and octave band spectra showing contribution to noise at 7 m (270°) from major sources in the Worthington 750-QTEX 73

45. Bar graph of 750-QTEX component source levels .. 74

46. Noise reduction due to acoustic enclosure on Worthington 750-QTEX, attenuation at 7 m, and transmission loss through side panel 77

47. A-weighted overall levels and octave band spectra showing estimate of radiated noise at 7 m 180° from Worthington 750-QTEX with an improved fan design 81

LIST OF TABLES

page

Table 1. Quieting techniques required for a 65-dBA compressor 34

2. Tonal noise generation at full power and speed . 40

3. Additional quieting of 750-QTEX for 65-dBA overall 84

4. British makers of portable air compressors 86

5. Major French manufacturers of portable air compressors 87

6. Balance of major European manufacturers 88

7. List price differences for standard vs quieted machines in Europe (all diesel-driven) 90

8. Chief competitors to American manufacturers 92

9. Consumer cost of quieting standard machines 96

10. Material costs used 100

11. Worthington 160 QT monorotor quieting costs 101

12. Atlas Copco VSS-170 super silenced model quieting costs 102

13. Worthington 750-QTEX "Blue Brute" quieting cost 103

14. Multipliers for list price comparison 104

15. Calculation of marginal cost of quieting (consumer) for Worthington 160 monorotor (gasoline-driven) 106

16. Calculation of marginal cost of quieting (consumer) for Atlas Copco VSS-170 Dd (diesel-driven $\leq$ 500 cfm) 107

LIST OF TABLES *(Cont.)*

page

Table 17. Calculation of marginal cost of quieting (consumer for Worthington 750-QTEX "Blue Brute" (diesel-driven > 500 cfm) 108

18. Quieting the Worthington 160 QT unit to 65 dBA . 110

19. Quieting the Atlas Copco VSS-170 unit to 65 dBA 111

20. Quieting the Worthington 750-QTEX unit to 65 dBA 113

21. Comparison of quieting costs for 65-dBA machines 114

22. Listing of performance parameters 118

1. INTRODUCTION

The Noise Control Act of 1972 requires the Environmental Protection Agency to regulate the noise of new products in several categories, including the construction industry. Section 6 of the Act calls for consideration of two sets of statistics in the regulation of noise: (1) the degree of noise reduction achievable by the application of the best available technology and (2) the cost of compliance.

Cost of compliance is the subject of BBN Report No. 2566c, the first BBN report on air compressors, published in September 1973, which discusses cost in terms of list price difference between standard and quieted units. In Report No. 2566c, cost refers to end-user cost, not manufacturers' cost of quieting.

In this report, BBN No. 2795b, we analyze the noise of three air compressors and assess the cost to manufacturers of quieting these compressors. To aid in identifying the most significant contributions to A-weighted dBA noise levels, we conducted all spectral analyses after processing the data with an A-weighted filter.

Three compressors and four engines are available in the air compressor industry. We report on the following combinations:

- The Atlas Copco VSS-170 Dd unit, a 170-cfm reciprocating-type compressor driven by an air-cooled diesel engine.
- The Worthington 160 QT, a 160-cfm sliding-vane compressor driven by a gasoline-fueled water-cooled engine.
- The Worthington 750-QTEX, a 750-cfm rotary-screw compressor driven by a water-cooled diesel engine.

All three are quieted units.

From research and testing on these units, we assess the costs in material and labor to air compressor manufacturers for quieting their products. The economic impact of quieting compressors can then be projected from estimated manufacturers' costs, rather than manufacturers' list price differences.

Sections 2, 3, and 4 discuss the individual characteristics of the three tested compressors. Included in each section is a noise-source diagnosis, an operational performance diagnosis, and estimated quieting costs. Included with quieting costs is the estimated cost of alternative designs for even quieter units.

Section 5 deals with foreign air compressors. After surveying major European air compressor manufacturers, we assess the noise control technology used on their units to meet stringent noise regulations. We also determine the price difference in Europe between "standard" units and "quieted" versions.

2. GASOLINE ENGINE-POWERED AIR COMPRESSOR

2.1 The Worthington 160 QT Air Compressor

The 160 QT compressor unit consists of a 65-hp gasoline engine driving an 8-vane compressor with a design output of 160 cfm at 100 psig. General views of the unit (Figs. 1 through 3) show that it is basically a conventional unit with added sound treatment and enclosures. The manufacturer developed this unit from the basic design by adding a fan inlet plenum, a larger exhaust muffler, an engine and compressor inlet muffler, and an enclosure at the front of the assembly. The air/oil tank, which surrounds the compressor, has been lagged (i.e., wrapped with sound insulation). A sheet steel plate encloses the bottom of the assembly. Figure 4 shows a diagrammatic view of the main design features; note that the engine and compressor casings are rigidly connected together as well as to the chassis frame. The unit is designed to be operated with the side doors closed, thus completing the enclosure of the internal noise sources.

2.2 Noise Aspects of the Design

We conducted tests to establish the contribution of individual sources to the total noise signature of the Worthington 160 QT. Significant sources were found to be: the compressor, the engine and its cooling fan, the exhaust outlet and muffler shell, and the air intake.

The engine and compressor assembly radiate noise directly, with the compressor assembly somewhat attenuated by the surrounding air-oil tank. In addition, since they are rigidly attached to the chassis and the shell of the machine, engine and compressor vibration is transmitted directly to the frame and outer sheet metal, which also vibrate and radiate noise.

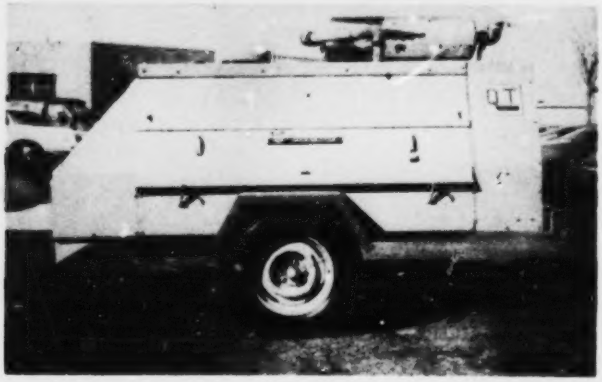


FIG. 1. GENERAL VIEW OF WORTHINGTON 160 QT

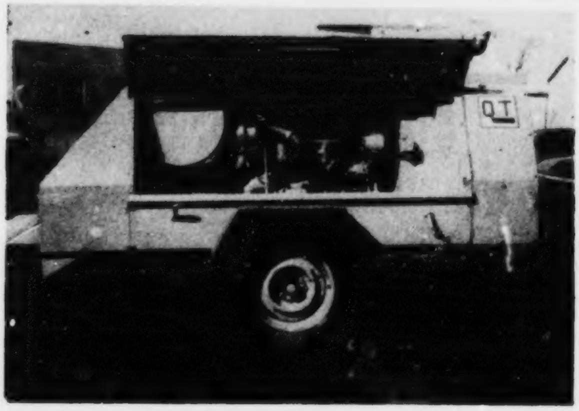


FIG. 2. WORTHINGTON 160 QT WITH SIDE CURTAIN OPENED

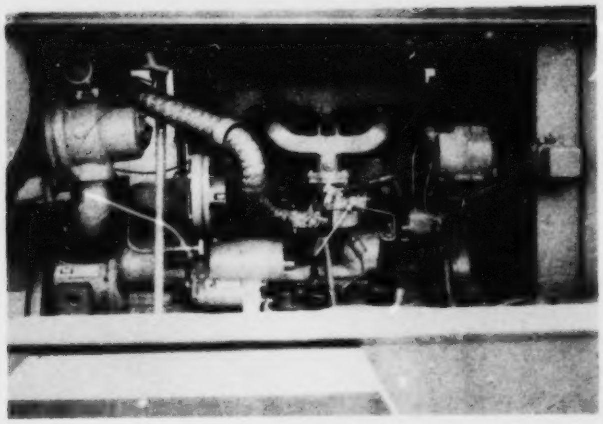


FIG. 3. CLOSEUP OF WORTHINGTON 160 QT ENGINE AND COMPRESSOR COMPARTMENT.

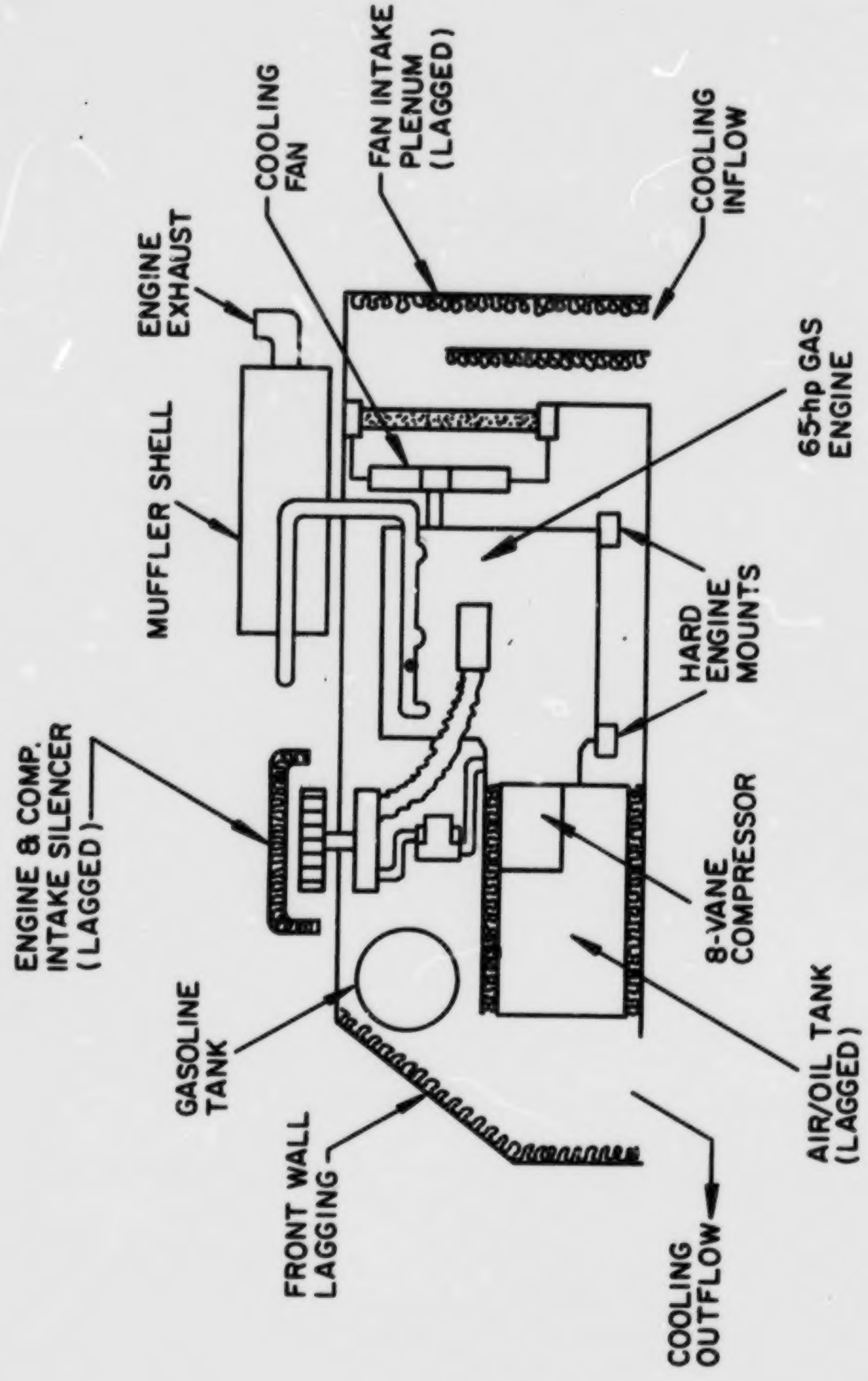


FIG. 4. WORTHINGTON 160 QT MAIN DESIGN FEATURES.

The engine cooling fan can exhibit strong broadband noise because of blade stall, together with leading- and trailing-edge turbulence. Uneven pressure distribution upstream and downstream of the fan may cause additional noise in discrete tones. (Non-optimal design of the fan shroud will permit flow recirculation at the blade tips.) These factors, in addition to generating noise, reduce efficiency of both the fan and the overall cooling system.

The engine exhaust and muffler arrangement produces noise because of the direct discharge; it can also radiate noise from the large muffler shell vibrating with the internal pressure fluctuations.

The air-intake system supplies the engine and compressor through a common air filter and silencer. The two air induction pressures thus combine to form a separate noise source.

2.3 Testing

2.3.1 Test program

For noise-source diagnosis, we isolated individual source contributions either by suppressing the other sources or, where possible, by running a source independently at its normal operating condition. The tests were all conducted at full engine speed (2400 rpm) with rated flow from the compressor.

Initial tests were conducted in the BBN semianechoic test facility to determine contributions from intake and muffler shell and the effect of opening the side door. Exhaust and compressor outlets were discharged from the room. Measurements were made in the geometric near field and at 5 m from the right side of the unit.

At first, tests were conducted with the unit completely covered with fiberglass and lead sheet. Subsequent tests progressively uncovered the muffler shell and intake apertures. Data were recorded on magnetic tape for later 1/3-octave band analysis. Figures 5, 6, and 7 show the test configurations.

Sound power and directivity tests were conducted in an empty parking lot with the microphones arrayed in a 10-point hemisphere of 7-m radius centered at the intersection of the ground plane and the geometric center of the machine.* Data were recorded on magnetic tape and later reduced in 1/3-octave bands and 4-Hz constant bandwidth spectra. In addition to the power level measurements, the test analysis provided preliminary source identification from the exhaust and internal sources of the machine.

We performed further tests in the empty parking lot to isolate the fan contribution to the noise signature. We ran the fan individually with an electric motor. The motor drive was on the left side of the unit; readings were taken at the 7-m right-side and 5-m rear positions. A notched wooden panel was used to replace the side door to provide a normal operating configuration of the unit and allow the rigging of an external belt drive. Figure 8 shows the fan-motor drive arrangement. Additionally, the fan intake plenum was removed to determine the effect of this silencing component on the rear position noise level.

2.4 Test Results

Directivity tests showed that the relative contributions of the individual sources at different measurement locations are dissimilar. For component source diagnosis, the standard position

---

*See the companion report (BBN No. 2795a) for a description of measurement locations.

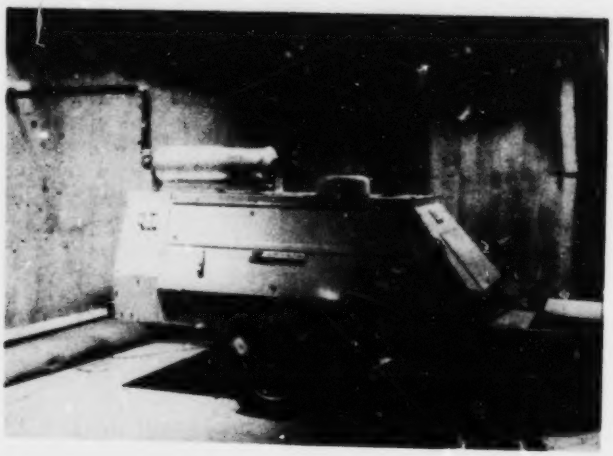


FIG. 5. COMPRESSOR INSTALLATION IN SEMIANECHOIC ROOM.

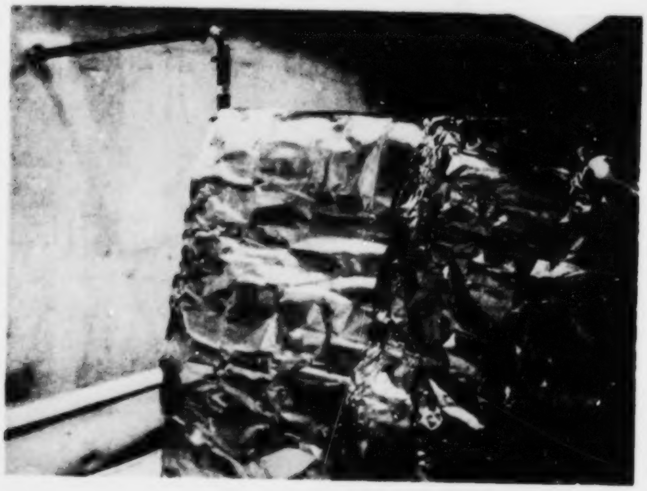


FIG. 6. TOTAL WRAPPING OF UNIT FOR INITIAL SOURCE DIAGNOSIS.

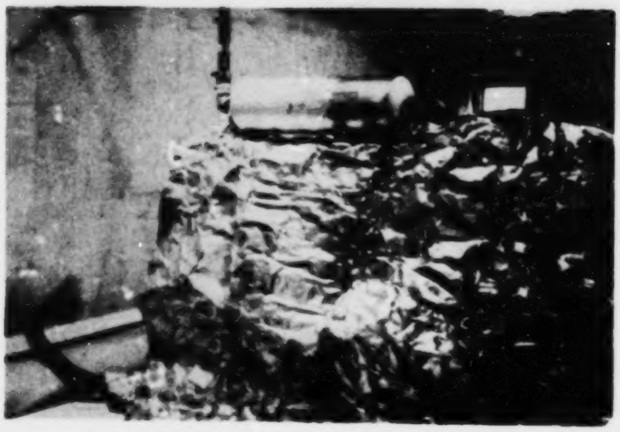


FIG. 7. MUFFLER SHELL UNCOVERED.

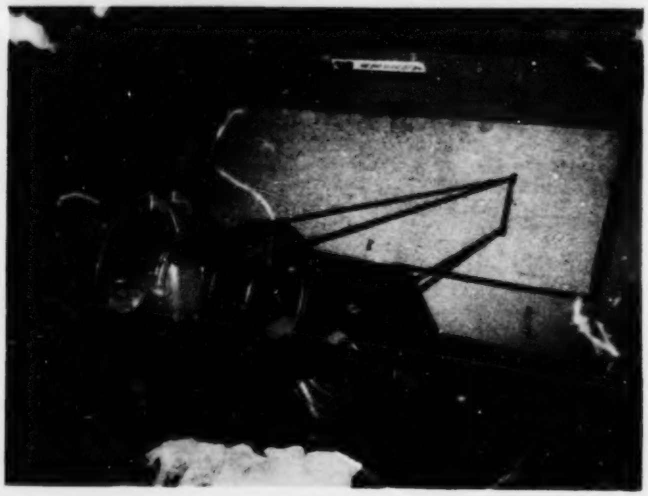


FIG. 8. FAN DRIVE MOTOR AND BLANKING PANEL.

at 7 m from the right side was chosen to be representative, since it is one of the cardinal points of the CAGI/PNEUROP survey (see Appendix D of BBN Report No. 2795a). Figure 9, a summary plot of all results obtained at the 7-m right-side location, shows the measured spectrum of the individual sources together with the composite levels in 1/3-octave bands* and total A-weighted level.

As shown in Fig. 9, the exhaust outlet is the major low-frequency contributor, and the fan and muffler shell are significant in the midbands. The intake is not a significant contributor in any frequency band, but does contribute to overall sound levels.

The discrete-frequency table listed with Fig. 9 identifies the pure tones responsible for the spectrum peaks based on the following measured speeds.

FULL POWER

Engine:

Speed = 2400 rpm = 40 Hz
Firing Frequency = 80 Hz (4 cylinders with a 4-stroke cycle)

Cooling Fan:

Rotation Speed = 3360 rpm = 56 Hz
Fan Blade Passing = 336 Hz (6 fan blades)

Compressor:

Speed = 2400 rpm (engine speed) = 40 Hz
Vane Frequency = 320 Hz (8 vanes)

---

*Note that the ordinate of Fig. 9 is labeled as 1/3-octave band *A-weighted* levels.

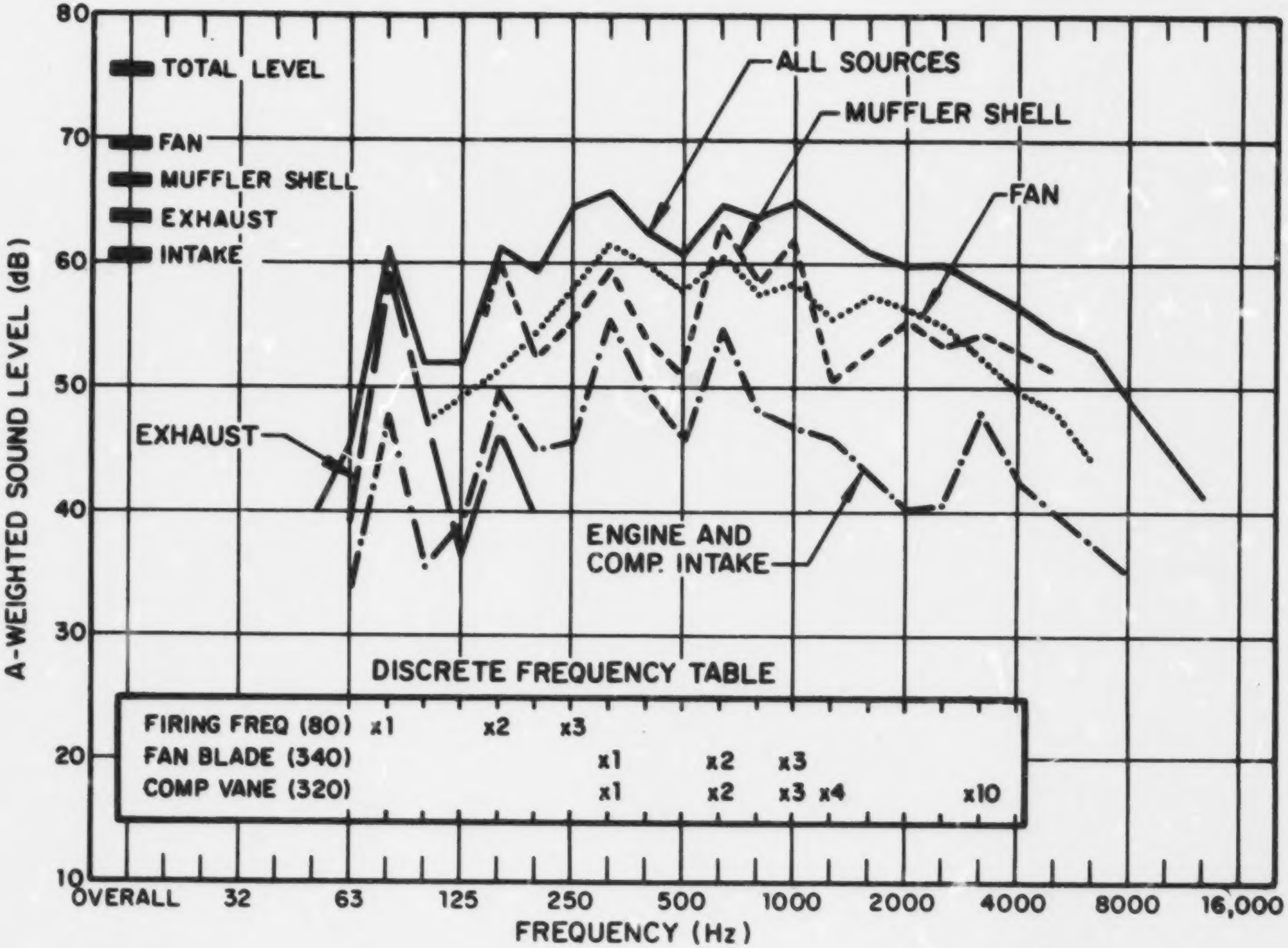


FIG. 9. INDIVIDUAL SOURCE CONTRIBUTIONS IN 1/3-OCTAVE BAND AND OVERALL LEVELS, MEASURED 7 m FROM RIGHT SIDE OF WORTHINGTON 160 QT COMPRESSOR.

The fan-blade and compressor-vane fundamental frequencies are separated by 16 Hz. This fact precludes their discrimination using analysis equipment with a 1/10-octave, or greater, bandwidth. Narrowband analysis is necessary to identify their individual contribution.

Figure 10, a bar chart of the individual A-weighted sound levels of major component sources, shows that the engine and compressor units are the major contributors to overall levels. Because of their close mechanical coupling, we could not separate these two sources. Nor, within this limited test program, could either unit be run individually under proper load conditions.

Results of the sound power and directivity tests are presented in Sec. 2, BBN Report No. 2795a. While the overall levels at each of the 10 locations did not vary by more than 1 dBA, markedly different spectra were obtained at these locations. Figure 11 illustrates this effect for two representative measurement points.

To aid in understanding the directivity of the sources, we obtained narrowband data below 2 kHz using a 4-Hz constant bandwidth filter. The resulting spectrum from one of the 10 locations is shown in Fig. 12, together with identification of the major peaks. An energy summation of the peak levels due to each source (i.e., engine firing, compressor vane, and fan blade) gives the major tonal contribution of each and some measure of the source directivities. Figure 13 presents the results of this analysis and shows the marked variation of each individual source, along with the fairly constant overall level.

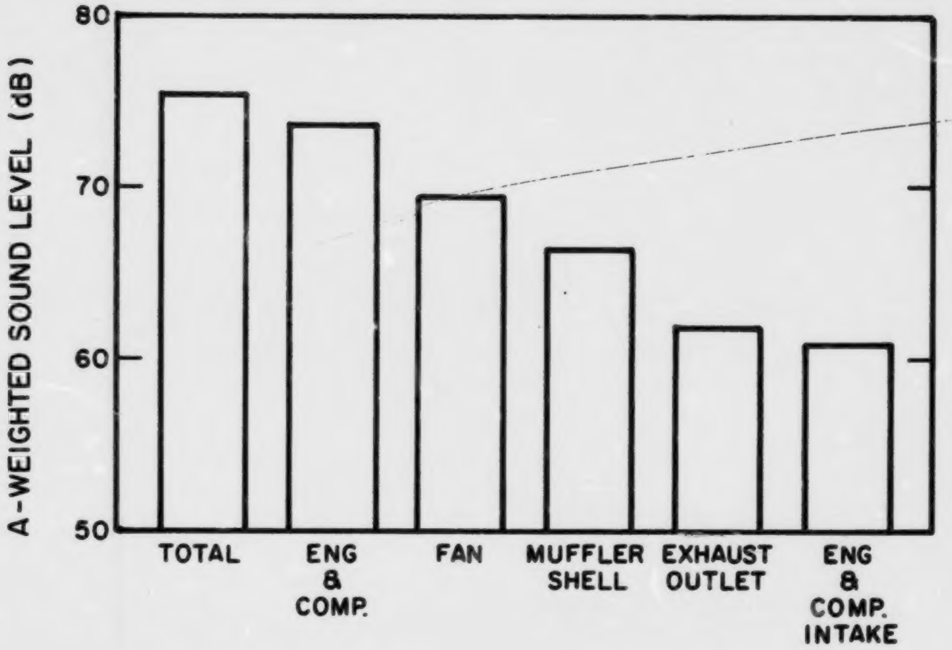


FIG. 10. SUMMARY OF SOURCE CONTRIBUTIONS MEASURED AT 7 m FROM RIGHT SIDE OF COMPRESSOR.

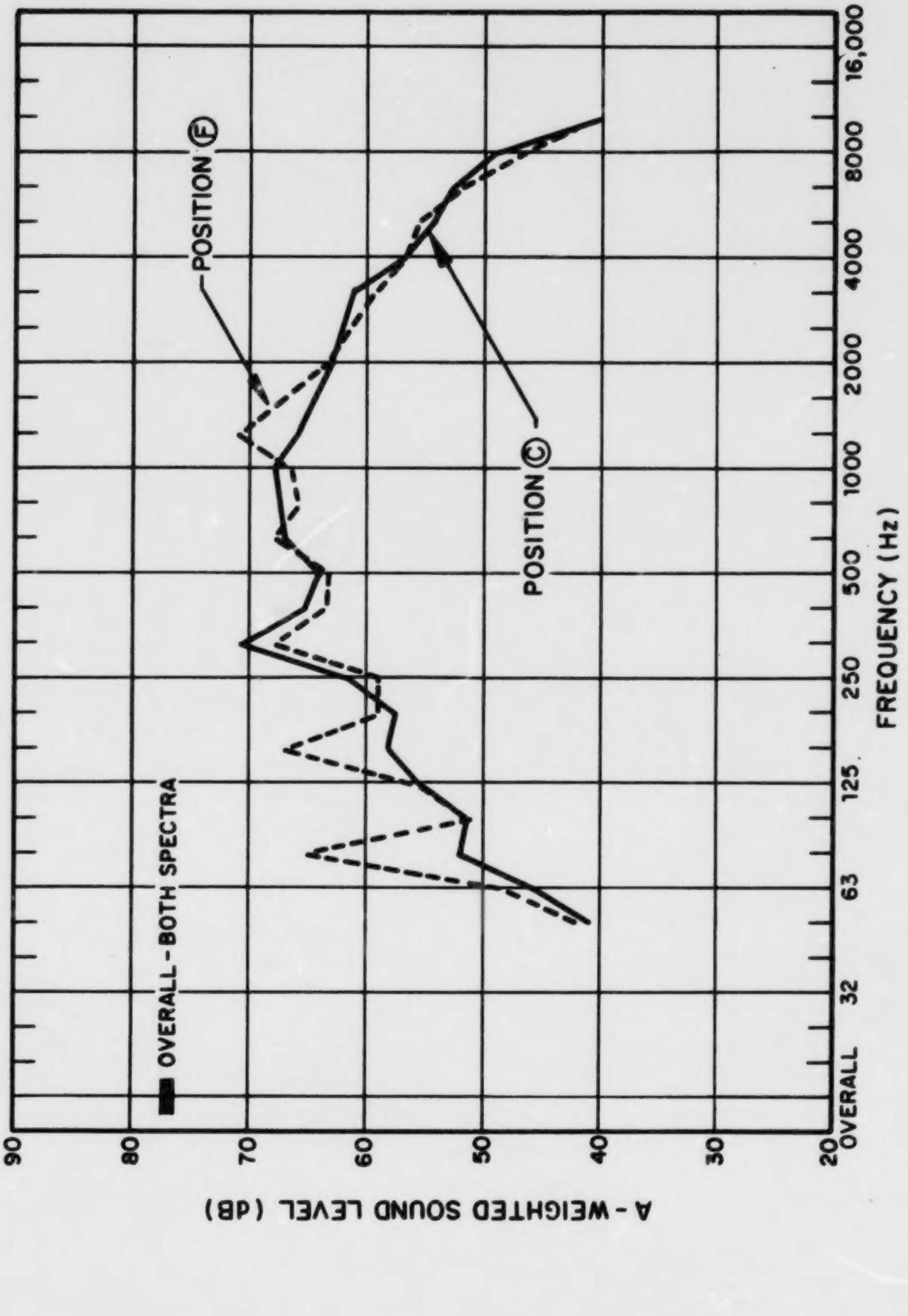


FIG. 11. TYPICAL VARIATION IN 1/3-OCTAVE BAND SPECTRA FOR TWO MEASUREMENT LOCATIONS ON 7-m RADIUS HEMISPHERE. FOR LOCATION OF POSITIONS, SEE TABLE IN REPORT NO. 2795a, TABLE 3.

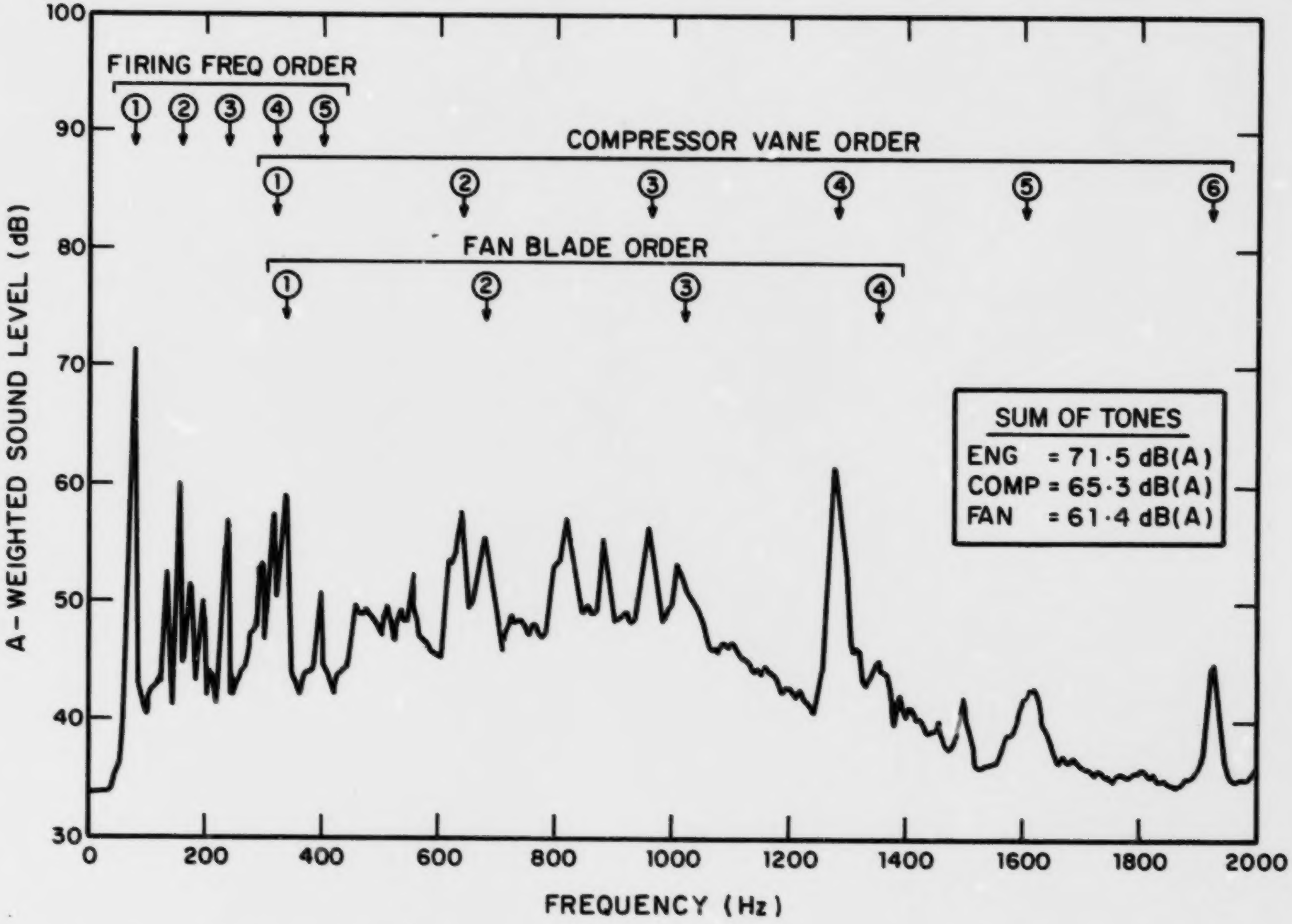


FIG. 12. CONSTANT BANDWIDTH ANALYSIS (4 Hz) OF SOUND MEASURED 6.4 m FROM LEFT SIDE OF FAN.

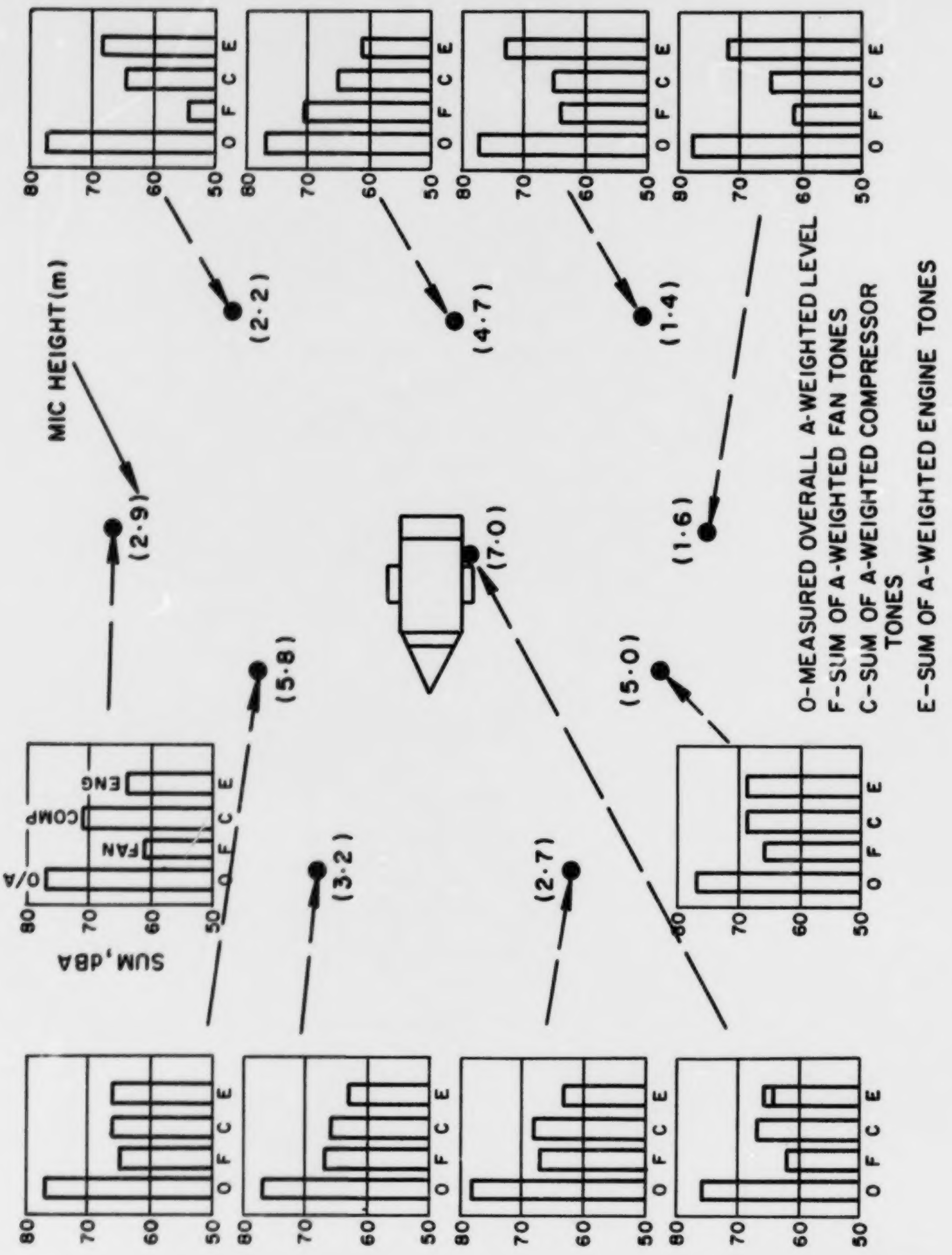


FIG. 13. PURE TONE DIRECTIVITY — FAN, ENGINE, AND COMPRESSOR.

2.5 Individual Noise Sources

2.5.1 Engine and compressor case-radiated noise

A test was run with the interior of the machine lined with fiberglass to reduce the reverberant buildup and the airborne path of engine and compressor case-radiated noise. No change in the right-side noise level was detected. This fact implies that the major contribution of the engine and compressor to exterior levels is a result of vibration of these components, transmitted to the structure through the rigid engine mounts. It was not possible to verify this fact directly by isolation of these units.

An attempt was made to run the compressor with the fan removed to measure engine and compressor noise alone. The result was almost immediate overheating, however, and stabilized operating conditions could not be obtained.

2.5.2 Engine cooling fan

The engine cooling fan was initially suspected to be significant noise source. Tests were performed with the fan driven externally to isolate its contribution to the overall level. Noise measurements were made 7 m from the right-hand side and 5 m from the rear surface, where the cooling-air intake is located.

A close-fitting wooden panel was installed on the left side of the unit with a small aperture to rig a drive belt from an external electric motor. First, the complete compressor was run to establish the effect of this panel on the noise level at the opposite side, the right-hand side. The electric motor was then rigged, as in Fig. 8, to drive the fan separately. Figure 14 shows the 1/3-octave band level results obtained from the two runs and compares them to the standard noise spectrum.

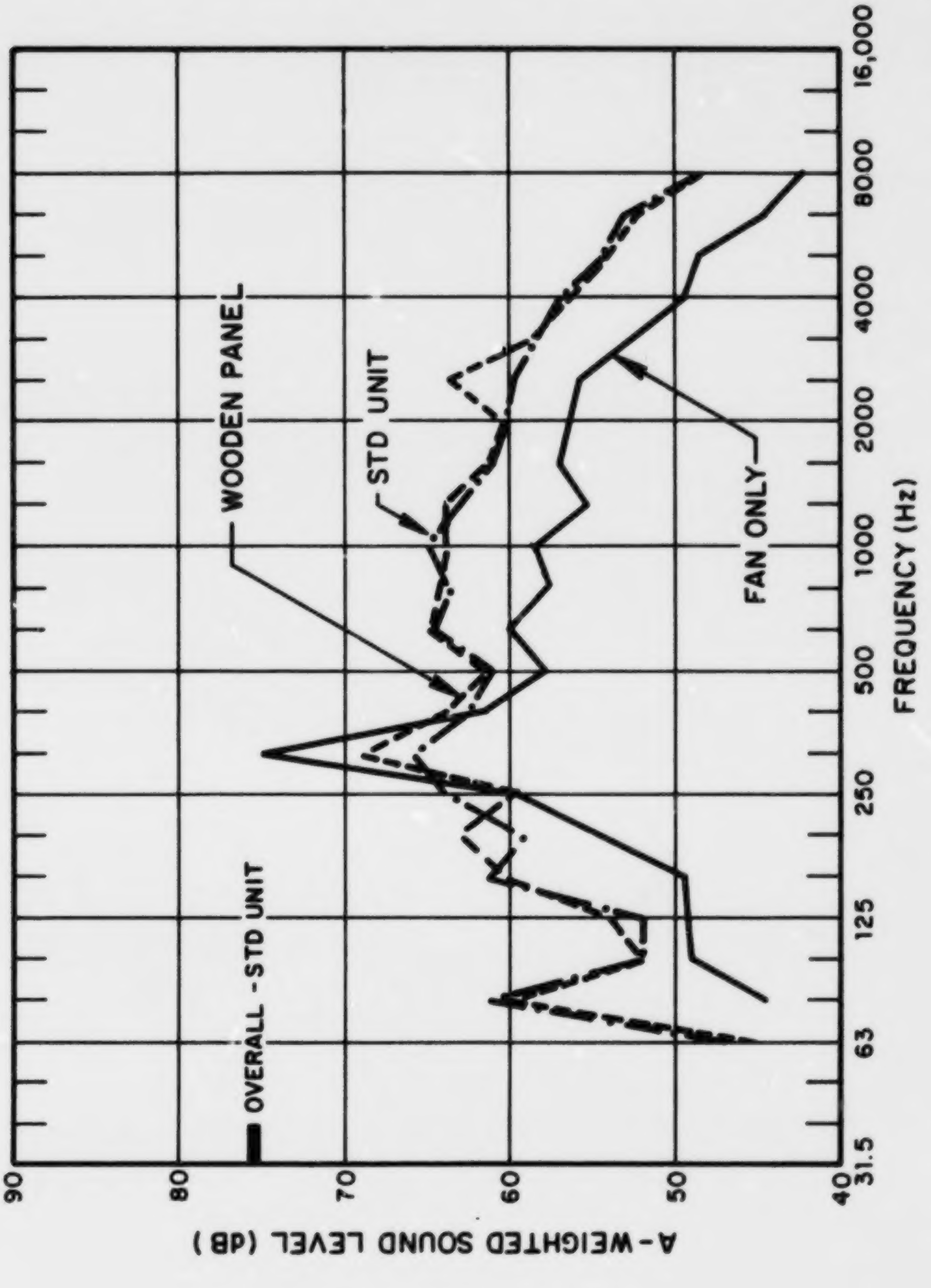


FIG. 14. 1/3-OCTAVE BAND AND OVERALL FAN NOISE LEVELS MEASURED 7 m FROM RIGHT SIDE OF COMPRESSOR.

This arrangement creates higher-than-standard levels at the fan blade-passing frequency of 336 Hz, which appears in the 315-Hz 1/3-octave band. The 3-dB increase at 336 Hz with the wooden panel is possibly caused by a localized bleed of air from the fan belt aperture just behind the fan. This airbleed could change the downstream pressure field, and increase a pure tone. The fan belt from the electric motor, which is close behind the fan blades, further changes the downstream air flow and may also be the cause of higher sound levels. When the fan is driven with an electric motor, there is a further increase of 6 dB. This test, then, appears to introduce at the blade-passing frequency a pure tone level greater than the normal configuration. The amplitude of this tone in the normal arrangement is obtained from narrowband spectra of the complete unit, i.e., 61 dB in the A-weighted 315-Hz 1/3-octave band. The test is considered valid for the broadband noise measurement since there is no significant change in the levels at other octave bands except at 3150 Hz. This change is possibly due to panel resonance or rattle from vibration at 10 times compressor-vane order. The fan noise contribution is presented in Fig. 9 as a composite of broadband noise from the individual fan test above 500 Hz and narrowband spectrum analysis of the complete unit at fan blade-passing frequency.

In order to assess the effectiveness of the fan-intake plenum on silencing, we removed the plenum component and conducted a repeat run. Figure 15 shows the obtained results, extrapolated to the 7-m CAGI standard distance, compared to the octave band readings taken during the CAGI/PNEUROP tests. The fan is the major contributor to noise levels at the rear of the unit. Removal of the intake plenum raises the fan noise level by 12 dBA and thus raises the total noise level by some 11 dBA.

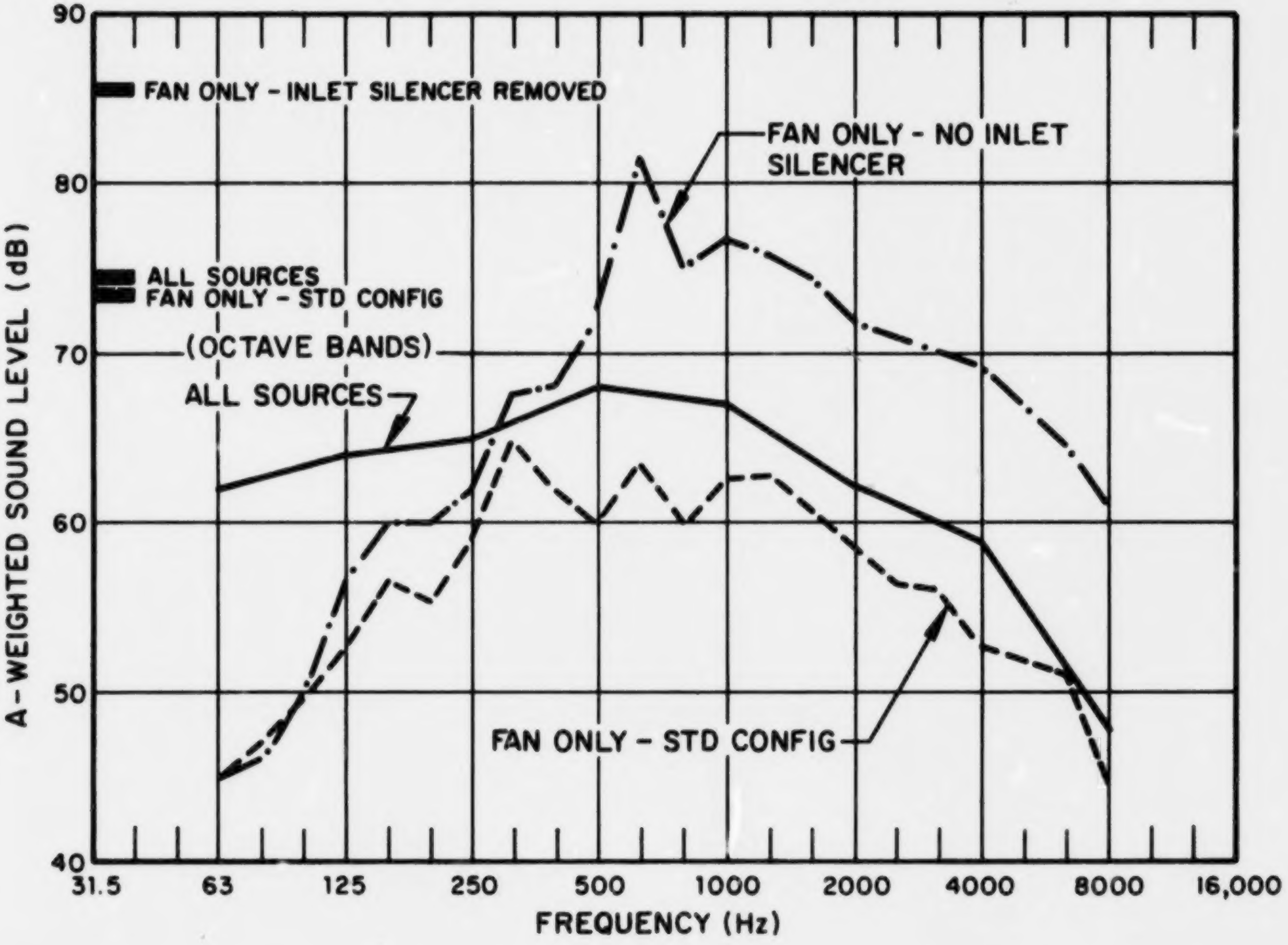


FIG. 15. 1/3-OCTAVE BAND AND OVERALL FAN NOISE LEVELS MEASURED 7 m FROM REAR OF COMPRESSOR.

Previous unpublished work at BBN suggests that this type of fan often operates in a stalled mode, with considerable tip recirculation due to inadequate shroud design. The proper fairing of a shroud to achieve a smooth transition from the rectangular radiator to the circular fan, at a given speed, can significantly reduce the noise due to tip recirculation. The associated efficiency increase allows-the fan speed to be reduced, yet the overall cooling system efficiency is maintained. On a similar fan installation, a 4-dBA observed reduction resulted from shroud redesign and a further 7-dBA reduction resulted from a decrease in fan speed. A reduction of 11 dBA in the fan noise suggests that we may dispense with the fan intake silencer except for reduction of direct engine and compressor-radiated noise.

Figure 16 shows a close-up of the fan and illustrates the type of shroud that permits air recirculation at the tip. During the fan-only noise tests, the air inlet was blocked to determine if the fan blade was operating stalled. The overall sound level and corresponding spectra did not change significantly, indicating normal fan operation is indeed in a stalled mode.

The fan is 18 in. in diameter and has six blades of 3.5-in. chord at an angle of 35° to the plane of rotation. International Harvester data (1971) show that a similar 20-in. fan gives a mean air flow of 5.9 lb/sec at 3360 rpm. Scaling by area results in a flow of 4.8 lb/sec for the Worthington fan at design speed. This is equivalent to 63 ft^3/sec or a mean velocity of 36 ft/sec through the fan.

The estimated fan inlet velocity distribution shown in Fig. 17a is based on BBN results for a similar fan design. Using velocity triangles, the air incidence angle onto the blade is

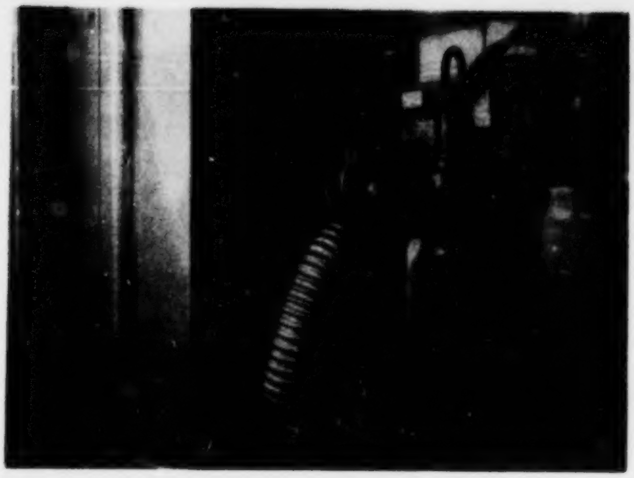
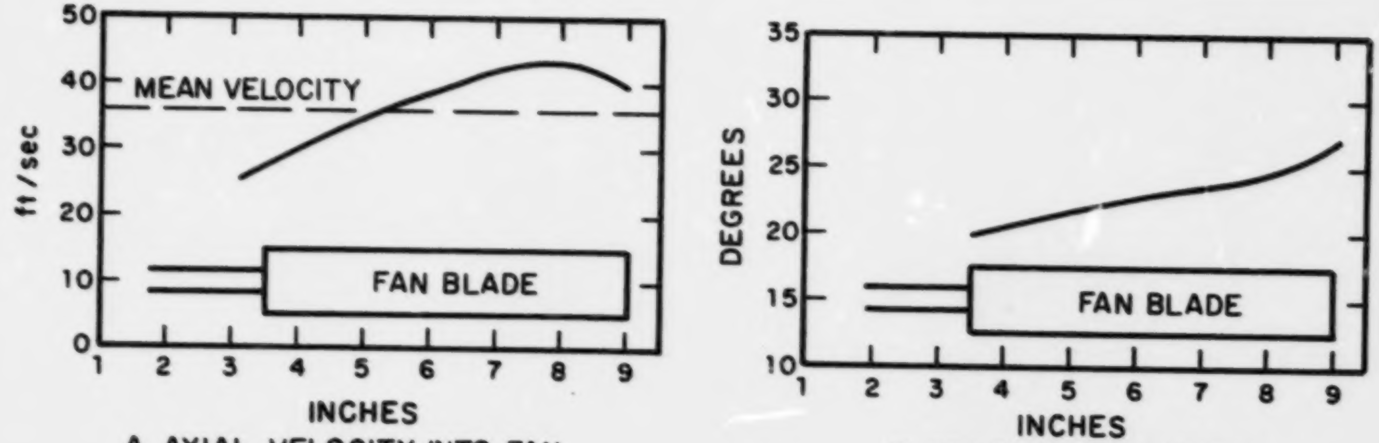
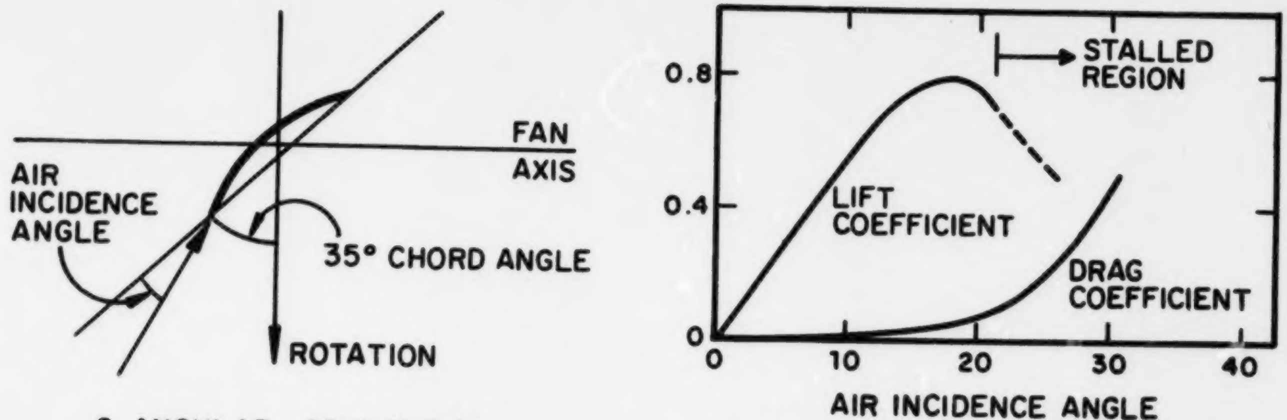


FIG. 16. ENGINE COOLING FAN AND SHROUD.



A. AXIAL VELOCITY INTO FAN

B. AIR INCIDENCE ANGLE



C. ANGULAR CONVENTION

D. TYPICAL LIFT AND DRAG CURVES

FIG. 17. COOLING FAN OPERATING CONDITIONS.

shown to be between 20° and 27°, as plotted in Fig. 17b. (Figure 17c shows the angular convention used.) Comparison with Fig. 17d shows that virtually all of the blade is operating stalled.

A reduction in the air incidence angle to prevent stall is achieved by applying variable radial twist to the blade. Unpublished BBN tests show that a 4-dBA reduction in noise is readily achievable, when the fan is unstalled and a correct blade twist is used.

Approximately 8 hp is required to drive the fan and water pump, as determined from the power fed to the electric drive motor. This amount would be reduced if the fan were designed to operate in a more efficient mode without blade stall.

In summary, a potential reduction of fan noise up to 15 dBA may be possible, through relatively simple shroud and fan blade redesign.

2.5.3 Exhaust and muffler noise

The prime component of the exhaust noise is at the firing frequency of 80 Hz. In all other 1/3-octave bands, the exhaust-related sound power is more than 12 dB below the 80-Hz component, as determined by a measurement 0.6 m from the pipe outlet. There is no major contribution at 80 Hz from any other source, including the engine inlet, as determined from results obtained in the semianechoic room.

Efforts to compute the sound power of the exhaust are hampered by significant ground reflections. Figure 18 presents the 80-Hz component, recorded during the sound power and directivity test, in terms of distance from the exhaust outlet. It is seen that

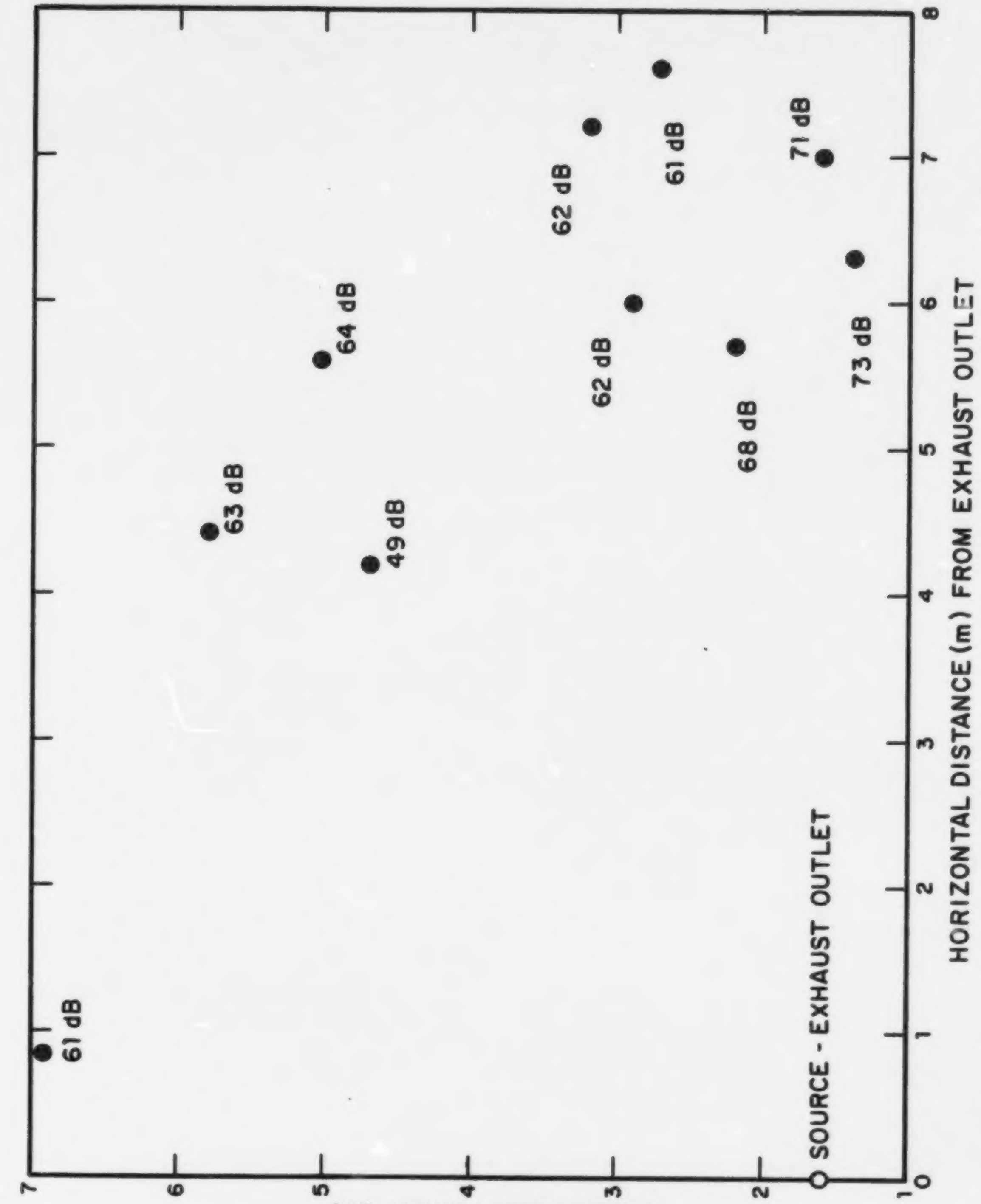


FIG. 18. FIRING FREQUENCY NOISE VARIATION WITH DISTANCE AND HEIGHT ABOVE SURFACE (A-weighted).

variations of up to 24 dB exist. In view of the fact that the exhaust is a point source and emits a tone, these variations are most probably due to interference between the direct and reflected paths.

Tests in the semianechoic room, with and without muffler lagging, show that the muffler shell generates an SPL at 5 m of 69.5 dBA, which, when extrapolated to 7 m, yields 66.5 dBA. The frequency spectrum derived from the muffler shell, displayed in Fig. 9, shows that the major contributions to overall A-weighted sound levels are in the mid- to high-frequency ranges.

Bender *et al.* (1974) show from tests on a truck that the muffler shell is excited both by mechanical vibrations transmitted along the exhaust pipe and by internal exhaust pressure fluctuations. The mechanical vibration path was more significant in this truck. Since the exhaust pipe used on the 160-cfm Worthington air compressor is connected solidly to the engine, we suspect the mechanical path to be important here, too. Therefore, a vibration isolator in the exhaust pipe would probably reduce the noise from the muffler shell.

2.5.4 Compressor and engine air intake

A common intake pipe from an air filter and silencer provides inlet air to both engine and compressor. Figure 19 shows the intake silencer; the inlet is the visible rectangular opening.

Nearfield tests in the semianechoic room established the sound power levels from both units as they are radiated from the intake. Figure 20 presents the nearfield results extrapolated to 7 m. The silencer attenuates the intake noise by 11-dBA overall. Tests were run with the intake silencer removed to determine the

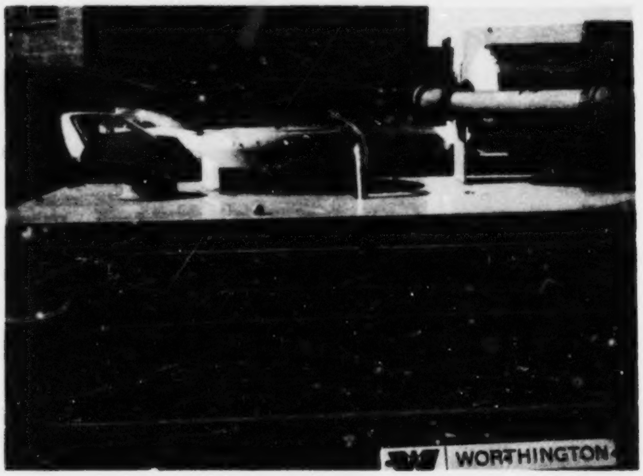


FIG. 19. COMPRESSOR AND ENGINE INTAKE SILENCER.

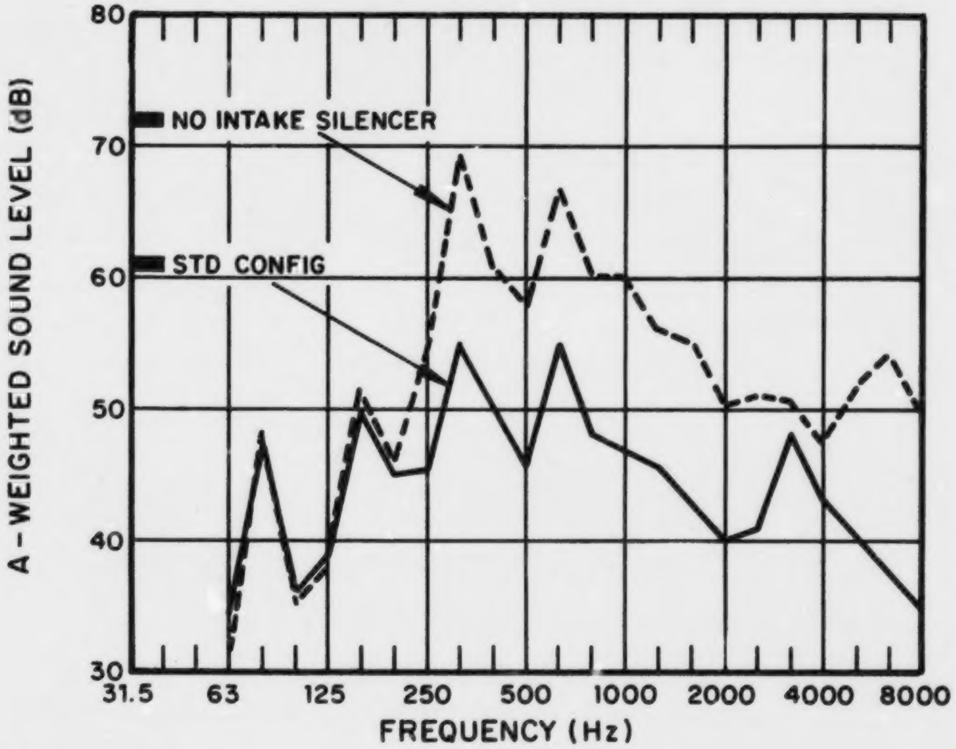


FIG. 20. ONE-THIRD OCTAVE BAND AND OVERALL ENGINE AND COMPRESSOR INTAKE NOISE LEVELS MEASURED 7 m FROM RIGHT SIDE OF COMPRESSOR.

individual engine and compressor contributions. It is clear that the major contributor to air intake-related noise is the compressor.

The intake at 15 dBA below the overall level (see Fig. 9) does not significantly contribute to the machine noise level.

2.5.5 Side-curtain effect

Tests were run with the side access doors both opened and closed. They were conducted in the semianechoic rooms; hence, the 80-Hz exhaust tone is absent. With the doors open, the overall noise level was 5-dBA higher (see Fig. 21). This is not necessarily a measure of the transmission loss of the doors, however, since with the doors open, fan operating conditions change as discussed previously. Moreover, the enclosure radiating surface is reduced.

2.6 Noise Control Techniques

2.6.1 Effectiveness of noise control techniques

The cooling-air intake-plenum silencer reduced the fan noise by 12 dBA and was thus judged effective. However, it is a heavy item and its location at the back of the unit tends to make the machine tip rearwards, particularly when the gas tank is empty. A better solution to the fan-noise problem would be improvement of fan design, eliminating the need for the silencer with its attendant weight and cost. Appendix A lists the costs associated with the cooling-air intake silencer.

The engine- and compressor-intake silencer was judged effective in providing 11-dBA attenuation; no major improvement is necessary in this area.

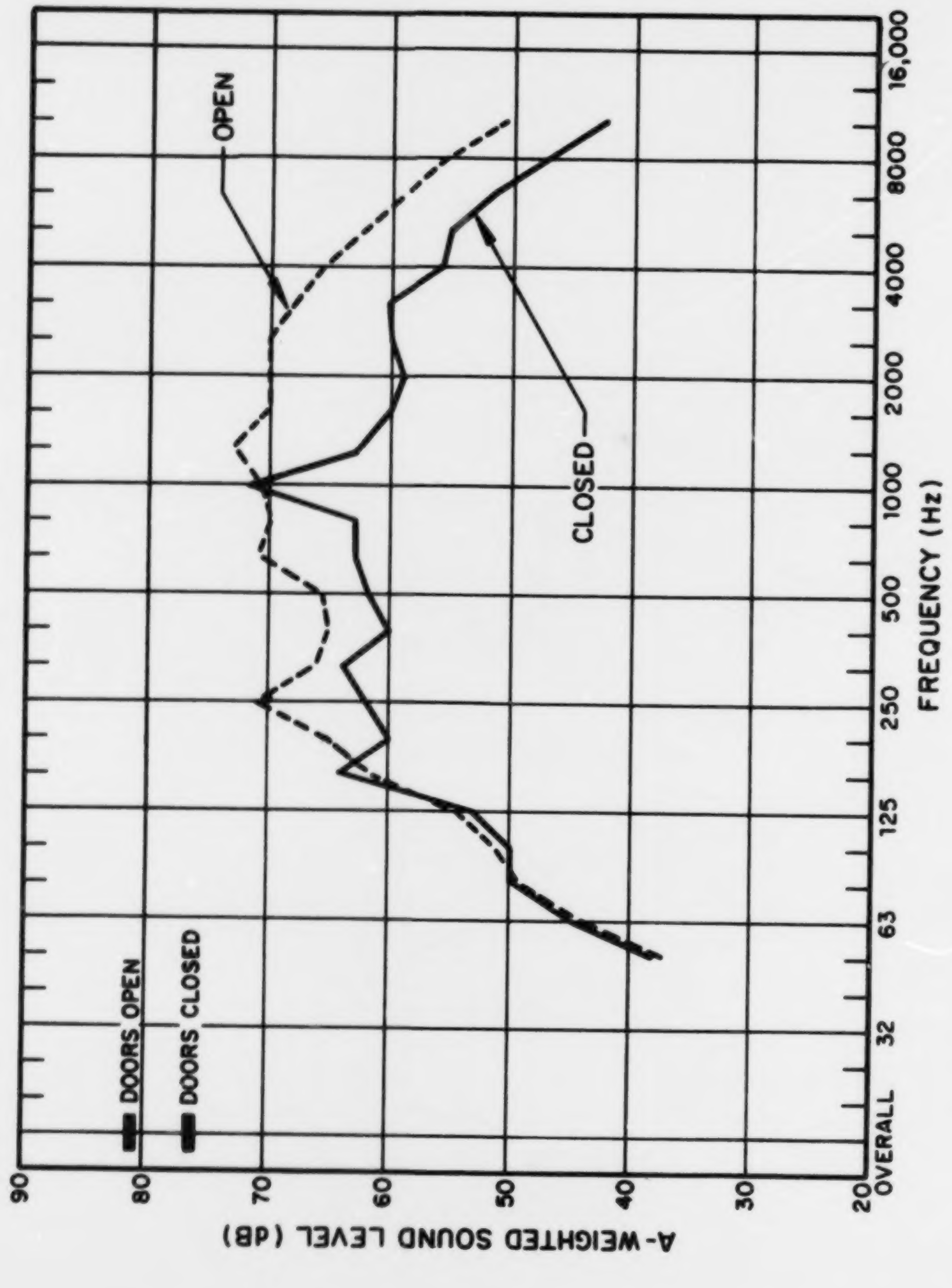


FIG. 21. EFFECT OF SIDE CURTAIN DOOR ON 1/3-OCTAVE BAND AND OVERALL NOISE LEVELS MEASURED 7 m FROM RIGHT SIDE OF COMPRESSOR.

The exhaust muffler exhibited shell radiation. Lagging of this unit and vibration isolation from the exhaust pipe would be beneficial if further quieting of the unit is necessary.

The major contributor to the mechanical noise of the unit is probably the compressor, judging from the number of tones generated by this component. Installation of vibration isolators between the engine-compressor unit and the chassis will reduce the structure-borne vibration transmitted to the outer sheet-metal covers.

2.6.2 Operational diagnosis

From the manufacturers' data, the effect of the noise control measures is a reduction in the air-to-boil* temperature from 116°F for the standard unit to 111°F for the silenced design.

Fuel consumption at 4.57 gal/hr is unchanged with silencing.

Increase in maintenance time for the silenced unit is primarily due to increased difficulty in engine and compressor oil changes, because of the presence of the bottom pan. Access to the sides of the unit remains unchanged. Removal of the radiator or compressor unit is slightly harder because of the sound treatment, but as these operations are assumed to be infrequent, they are not serious problems.

2.7 Further Noise Reduction Programs

A reduction in the noise level at 7 m to 65 dBA would require some significant engineering redesign. When the unit was completely covered by lead sheet and fiberglass (as illustrated in Fig. 6), a 7-m level of 69 dBA was measured.

---

*Air-to-boil (ATB) is an engineering term used to indicate the ambient air temperature where continuous operation of the compressor would result in a bulk water temperature in the radiator top tank of 212°F.

Presently, the major noise source is engine- and compressor-induced structural vibration, at 74 dBA. A rated level for the unit of 65 dBA requires a reduction for this source of noise to approximately 60 dBA, which is feasible with vibration isolators. With this reduction, however, the direct casing-radiated noise from these units may require additional attenuation by adding acoustic absorptive material with included lead or leaded vinyl sheet to the side access doors.

The present level of fan noise of 69.5 dBA may be reduced about 11 dBA by shroud redesign and changes in both the fan speed and its blade twist.

A 10-dBA reduction in the muffler shell radiation is achievable by lagging with fiberglass and sheet metal, and vibration-isolating from the exhaust pipe. Total noise level would then be about 56 dBA.

The exhaust level of 62 dBA is reducible by the addition of a properly selected second muffler, or by substitution of the existing muffler with a different muffler having better attenuation characteristics. The engine and compressor intake noise could also achieve some additional noise reduction by use of a more refined intake silencer. Reduction of both of these noise sources to at least 57 dBA will be required to meet the 65-dBA overall goal. A summary of sample required source levels is given in Table 1.

The extensive extra treatment required suggests that reduction of noise generation at the source would be beneficial. In addition to redesign of the fan, consideration should be given to the use of a quieter compressor design, such as a reciprocating or rotary-screw type.

TABLE 1. QUIETING TECHNIQUES REQUIRED FOR A 65-dBA COMPRESSOR

Noise Source	Current Level	Target Level	Method
Engine and compressor casings	74	60	Vibration isolation plus increased transmission loss through side doors
Engine cooling fan	69	58.	Shroud redesign Blade twist Reduced fan speed
Muffler shell	66	56	Lagging of shell
Exhaust noise	62	57	Additional muffler
Engine and compressor intake	61	57	Redesigned silencer
TOTALS	76*	65	

The component costs for noise reduction are presented in Appendices A and B.

---

*Measured current level is 75 dBA. Component source levels were obtained independently and component sum may include other sources twice for higher overall level.

3. DIESEL ENGINE-POWERED AIR COMPRESSOR: 500 cfm OR LESS

3.1 The Atlas Copco VSS-170 Dd Compressor

The Atlas Copco Super Silensair (VSS-170 Dd) is a diesel engine-powered compressor, which supplies 170 cfm of free air at 100 psig and has a maximum working pressure of 125 psig. The reciprocating-type compressor is a 2-stage, 2-piston unit with an intercooler; the diesel engine is a Deutz 4-cylinder, 4-stroke-cycle air-cooled engine rated at 47 hp at 1750 rpm. Two cooling fans are used. One, a ducted axial fan, is belt-driven from the engine crankshaft to provide a flow of cool air over the engine. The other is a centrifugal-type fan attached to the compressor input shaft. This fan draws cool air into the enclosure and through the compressor intercooler. Figure 22 shows the Super Silensair air compressor and the standard Atlas Copco air compressor. Note the difference in the external sheet metal covers for the two units.

During initial engine start, a clutch disengages the compressor from the engine for easier rotation by the starting motor. This feature allows the engine to be operated alone, without the compressor. Thus, engine and compressor noise can be separated, rather than combined, as was necessary for the Worthington 160 QT compressor (Sec. 2).

3.2 Testing

Noise source diagnostic measurements were made both in a semianechoic chamber and outdoors. The test configuration used in the semianechoic chamber is shown schematically in Fig. 23

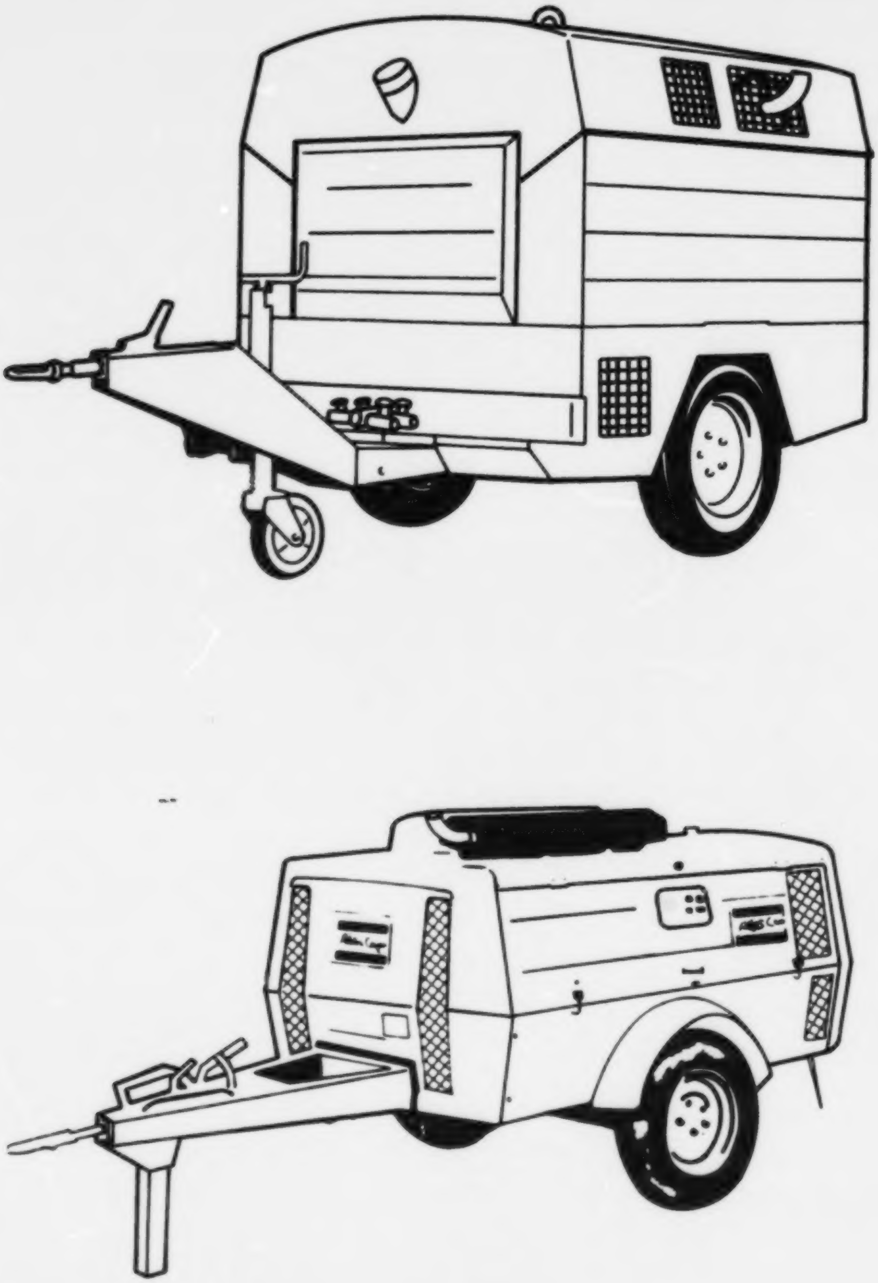


FIG. 22. ATLAS-COPCO AIR COMPRESSORS: ON TOP, SUPER SILENSAIR; BELOW, STANDARD MODEL.

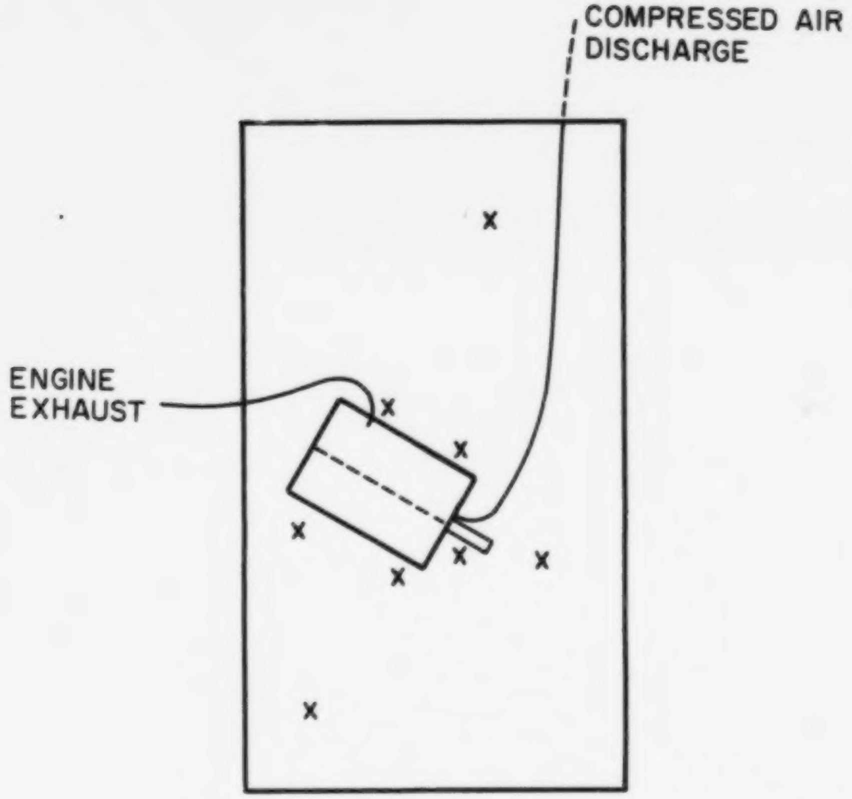


FIG. 23. ANECHOIC CHAMBER WITH COMPRESSOR INSTALLED SHOWING MEASUREMENT POSITIONS.

and pictorially in Fig. 24. Acoustic measurements were made at 0.3 m and 3.7 m from the sides, and at 0.3 m and 2.1 m from the front of the unit. The compressor was operated at full-rated power, with the engine exhaust and compressed-air discharge piped out of the chamber.

Three sets of measurements of compressor noise were made: (1) with the enclosure closed and latched, (2) with the enclosure fully opened, and (3) with the enclosure again closed and latched, after sound-absorbing fiberglass had been placed around the engine and compressor.

Outdoors, we characterized the noise of the compressor by using a 10-point 7-m hemispherical array. We performed the ISO-2151-1972 measurements for comparisons to existing data, and added an additional overhead measurement point: 7 m above the top of the compressor. The data were analyzed in 1/10-octave bands, in octave bands, and in dBA while the compressor was being operated. At the same time, we made tape recordings for later analysis, with A-weighting in 1/3-octave bands.

3.3 Ranking of Noise Sources

To determine the relative sound levels of the compressor components, we examined the spectral noise signature of the complete unit for tonal noise sources, placed the microphone adjacent to certain point sources such as the exhaust outlet, and operated the engine separately from the compressor.

Expected tonal noise generators and their calculated frequencies at full power and speed are listed in Table 2. Note that many have the same frequency. Examination of the A-weighted

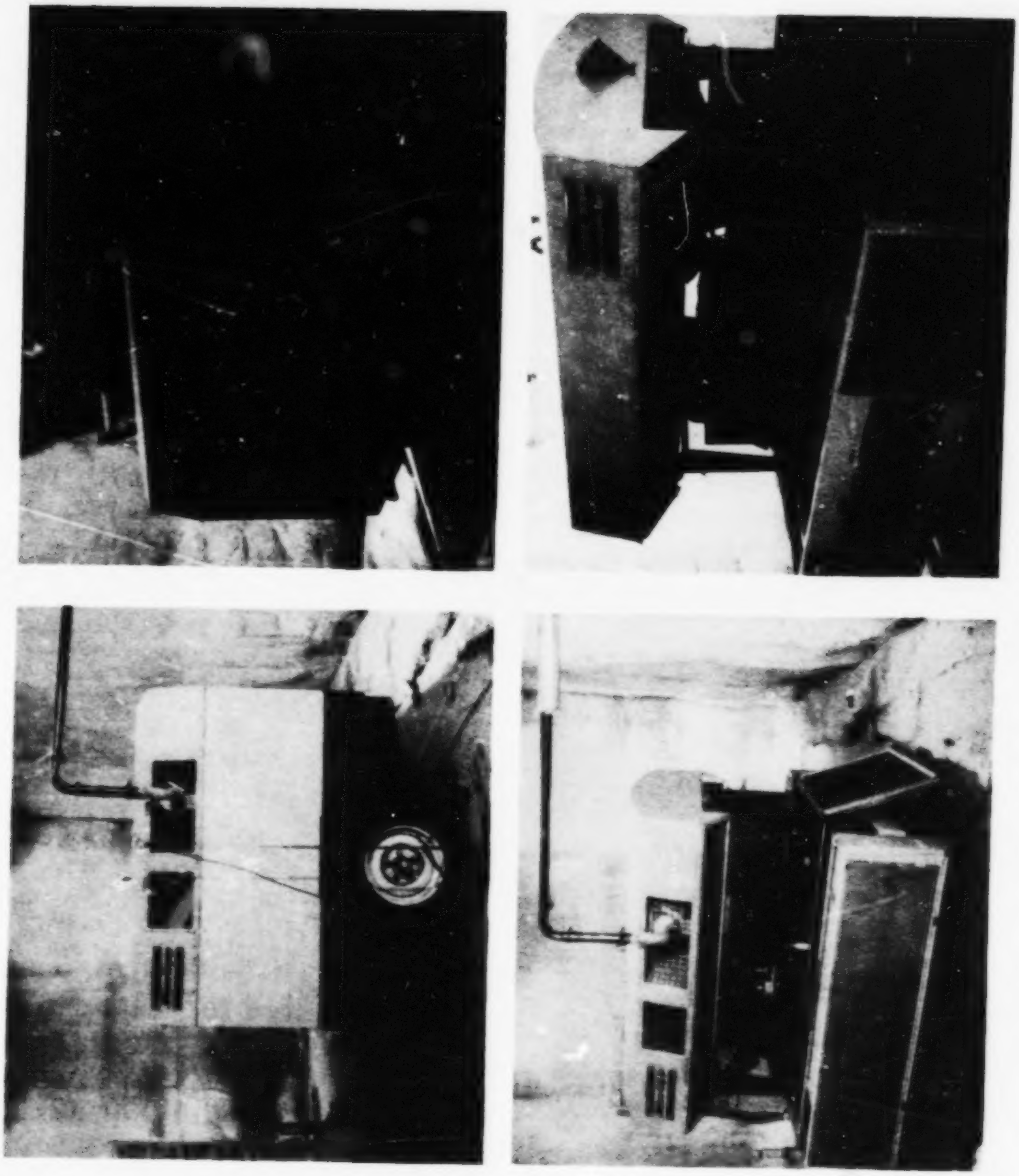


FIG. 24. FOUR VIEWS OF TEST CONFIGURATION USED IN SEMIANECHOIC CHAMBER.

1/10-octave band spectra for the unenclosed unit (Fig. 25) discloses tones significant to the overall A-weighted sound level at the engine-firing frequency (58 Hz) and its harmonics. Comparison of the relative magnitude of these spectral components at the cardinal 1-m distances around the unit with the compressor both on and off indicates that the harmonics are generated primarily by the compressor. The two fans do not seem to generate significant tones at their respective blade-pass frequencies.

TABLE 2. TONAL NOISE GENERATION AT FULL POWER AND SPEED

Engine rotation rate – 1750 rpm = 29 Hz
Compressor rotation rate – 1750 rpm = 29 Hz
Engine firing frequency – 58 Hz
Compressor pumping frequency – 29 Hz
Engine cooling-fan fundamental – 87 Hz
Engine cooling-fan blade-pass frequency – 1750 Hz
Intercooler fan fundamental – 29 Hz
Intercooler fan blade-pass frequency – 1400 Hz

Placing the microphone adjacent to the exhaust outlet and extrapolating the level to 7 m provides the estimated relative source level for exhaust-outlet noise. When exhaust-outlet noise levels are compared with the semianechoic room results where the exhaust gases are piped away, exhaust outlet noise is seen to consist of tones at the fundamental firing frequency and the first harmonic. The extrapolated 7-m level for exhaust outlet noise is 60 dBA.

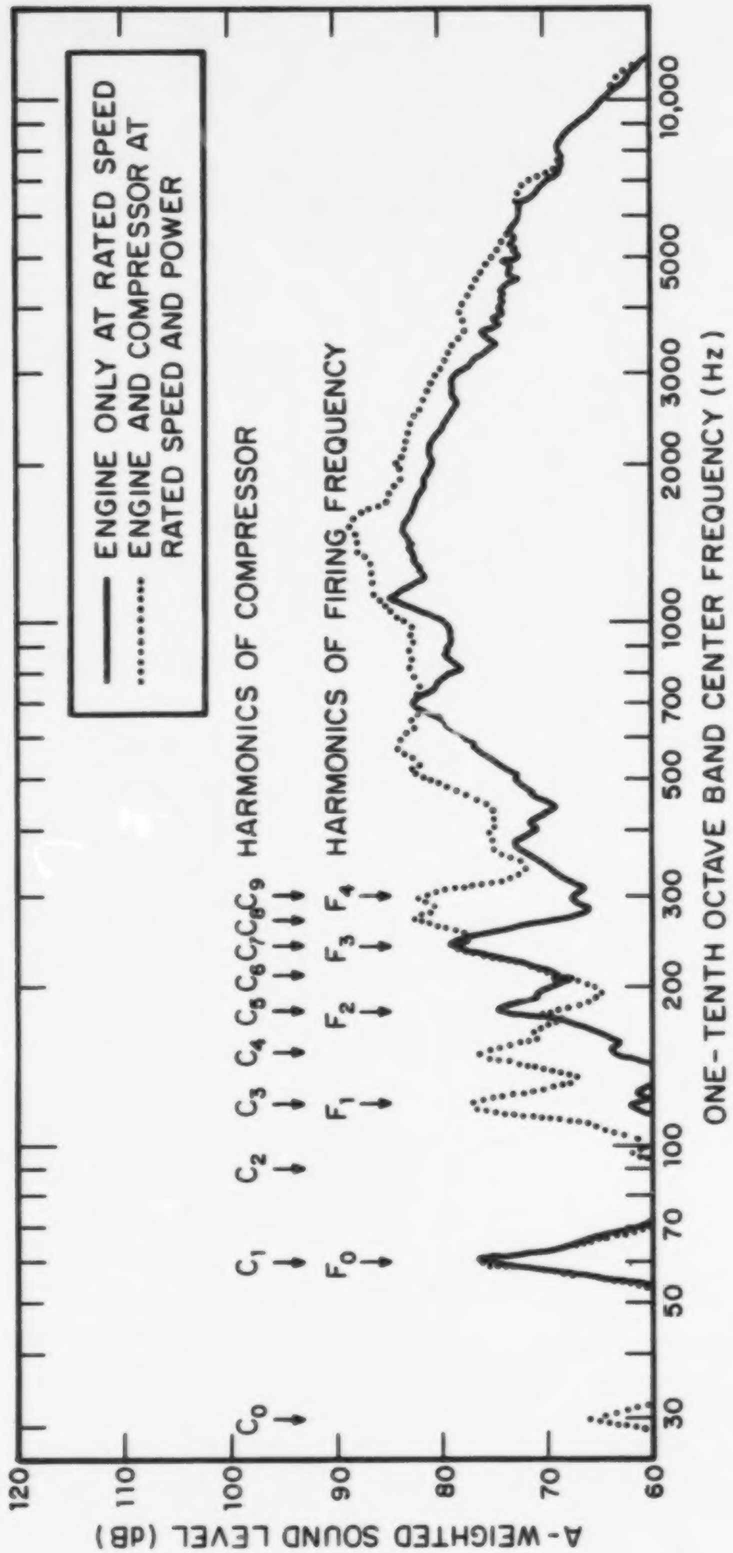


FIG. 25. SPECTRAL MEASUREMENTS OF UNENCLOSED UNIT AT 1 m.

When the microphone is positioned adjacent to the air intakes for the engine and compressor, with and without filters in place, the result shows that these sources do have tonal noise components at the compressor pumping frequency (29 Hz) and engine-firing frequency (58 Hz) and their harmonics. Because many of the frequencies are identical, the exterior radiated noise could not be separated. The estimated overall source level of both the engine and compressor intake, therefore, is 61 dBA when extrapolated from the close-in measurement.

The engine is automatically disengaged from the compressor during and after initial startup. When the engine is ready for loading, a button is pushed to engage the clutch and couple the compressor to the engine output shaft. The intercooler fan runs only when the compressor is operating. The engine and its cooling fan may thus be operated separately from the compressor. By blocking the governor arm, the engine is run at rated speed without loading. Engine noise, however, is strongly speed-dependent, and this procedure thus yields a reasonable approximation to the engine-related noise from block vibration and direct radiation. Fan noise is entirely speed-dependent and will be the same whether the engine is loaded or not. Figure 26 gives the results of sound tests at a position 7 m from the right-hand side of the machine. From these results, we estimate the source levels from the engine to be 63 dBA at a typical 7-m distance.

The compressor contributes a level of about 64 dBA, mainly from the strong radiation of tonal harmonics of the fundamental rotational rate. The contributions of both cooling fans are estimated to be 63 dBA from tests in which glass fiber bats were used to quiet the airborne path of acoustic radiation through the enclosure.

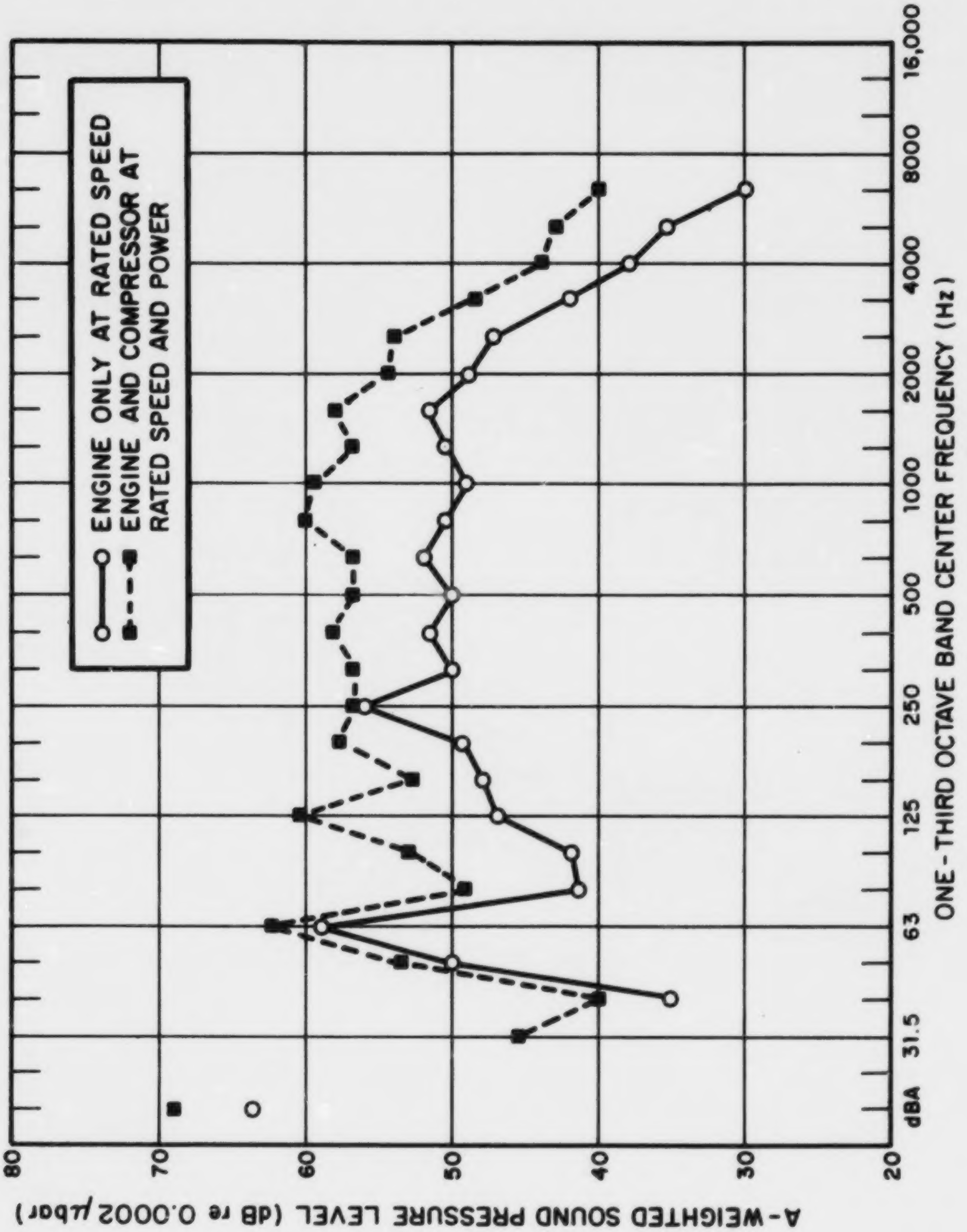


FIG. 26. MEASURED SOUND LEVEL AND A-WEIGHTED SPECTRUM AT 7 m FROM RIGHT SIDE.

The results of all these tests are summarized in Fig. 27, a bar graph indicating the relative contributions to overall levels.

3.4 Noise Control Techniques

Noise control techniques used to quiet the Super Silensair relative to the standard unit were most effective in the mid-frequency range, between 500 and 8000 Hz. The standard Atlas Copco compressor is equipped with a diesel engine-exhaust muffler (see Fig. 28), an engine-intake silencer, and a compressor-intake silencer to control the low-frequency noise (30 to 125 Hz). They are also used in the Super Silensair mode.

Mid-frequency silencing is accomplished by using an enclosure with foam-type acoustic absorptive material on the side walls and end doors (see Fig. 29). The enclosure has built-in ducting for the engine cooling air, as well as the compressor intake and cooling air (see Figs. 30 and 31). Cooling air exhausted from the diesel engine and the compressor and intercooler is ducted through another part of the enclosure before being discharged (see Fig. 32). These ducts are primarily effective in blocking line of sight or direct radiation from the engine and compressor to the outside. Figure 33 gives the measured noise reduction provided by the enclosure. Note the effect of added absorption in quieting the critical frequency range, 400 to 4000 Hz. The enclosure reduces the overall noise level of the unit by 11 dBA. The resulting spectrum with A-weighting is very flat from 30 to 2000 Hz.

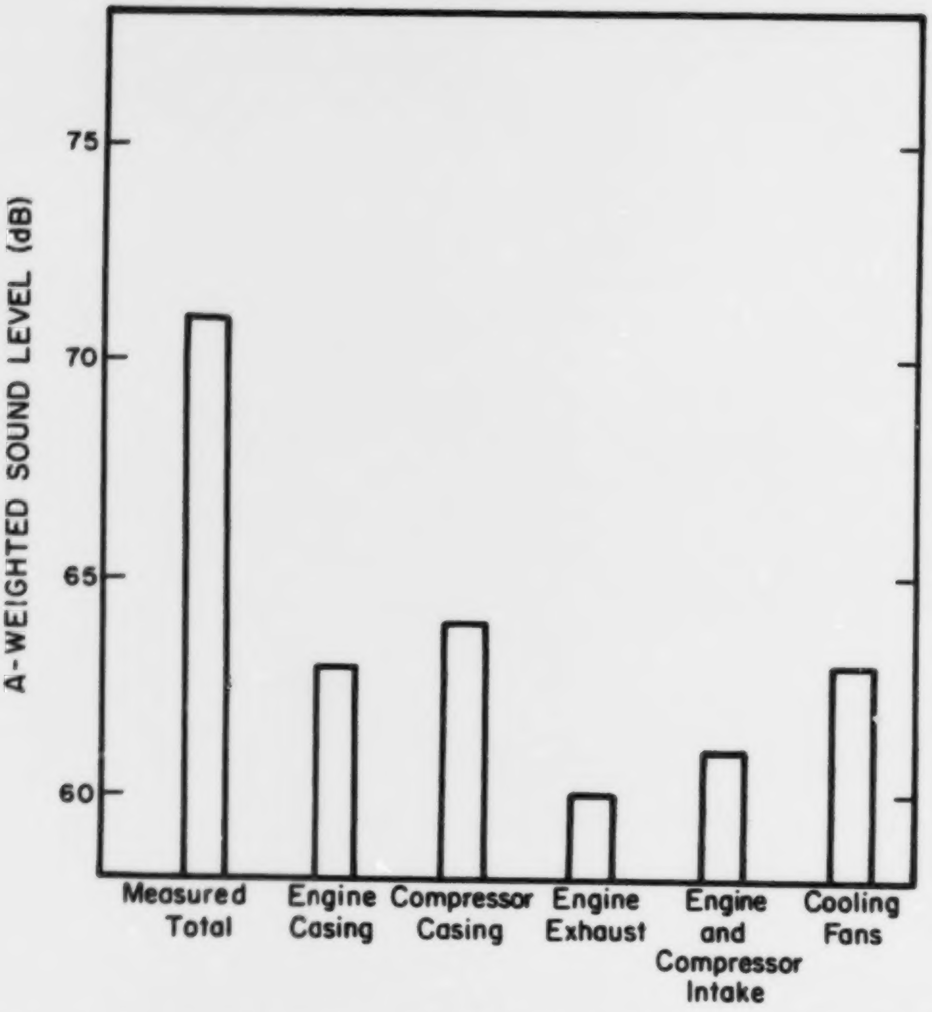


FIG. 27. CONTRIBUTIONS TO dBA ISO-2151-1972 SOUND LEVEL AT 7-m DISTANCE.

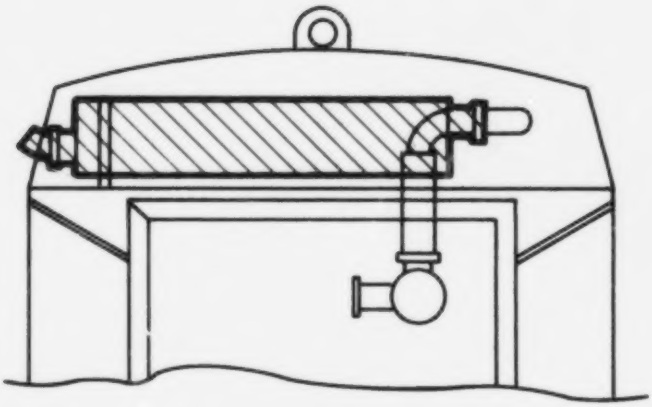


FIG. 28. EXHAUST SILENCER.

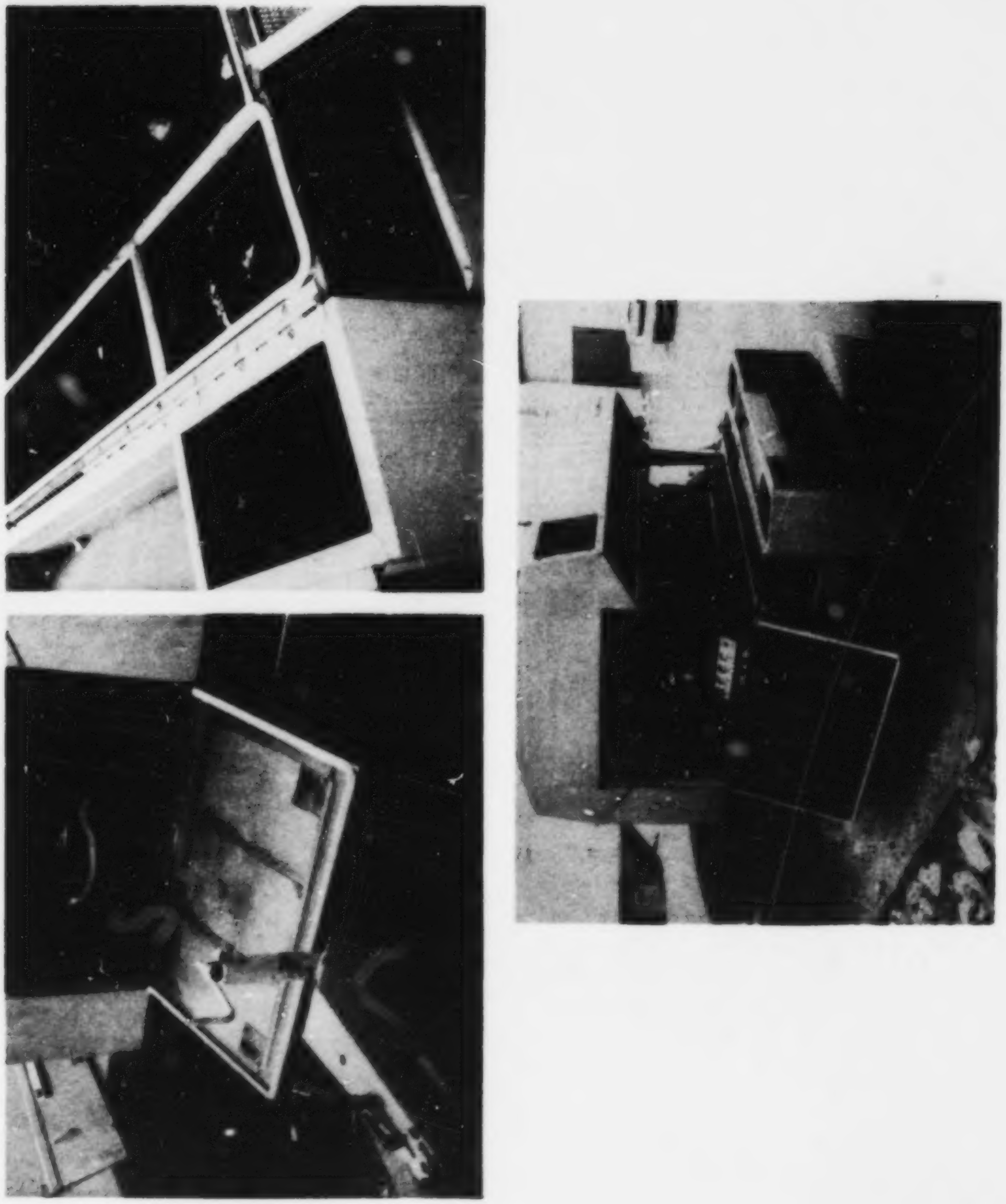


FIG. 29. VIEWS OF ACOUSTIC ENCLOSURE: SIDE WALLS AND END DOORS.

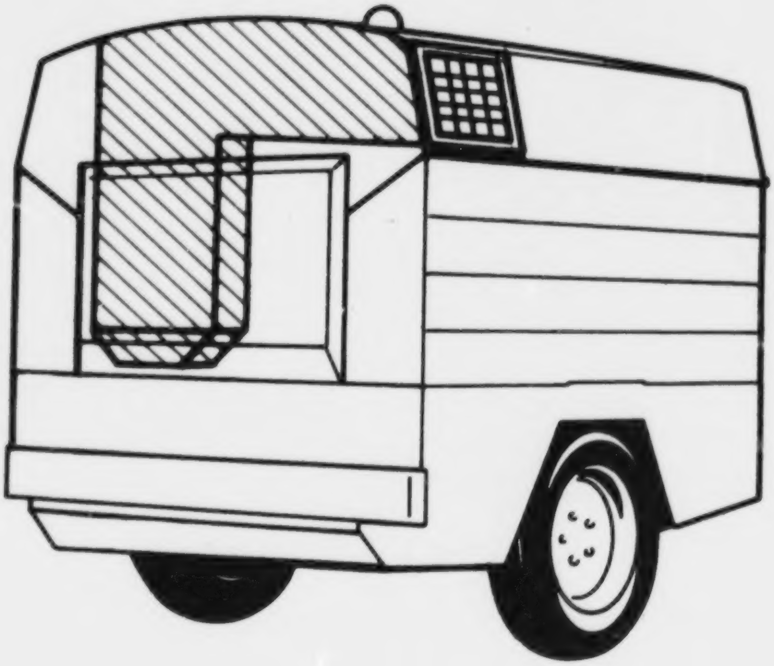


FIG. 30. ENGINE INTAKE AND COOLING AIR DUCTS.

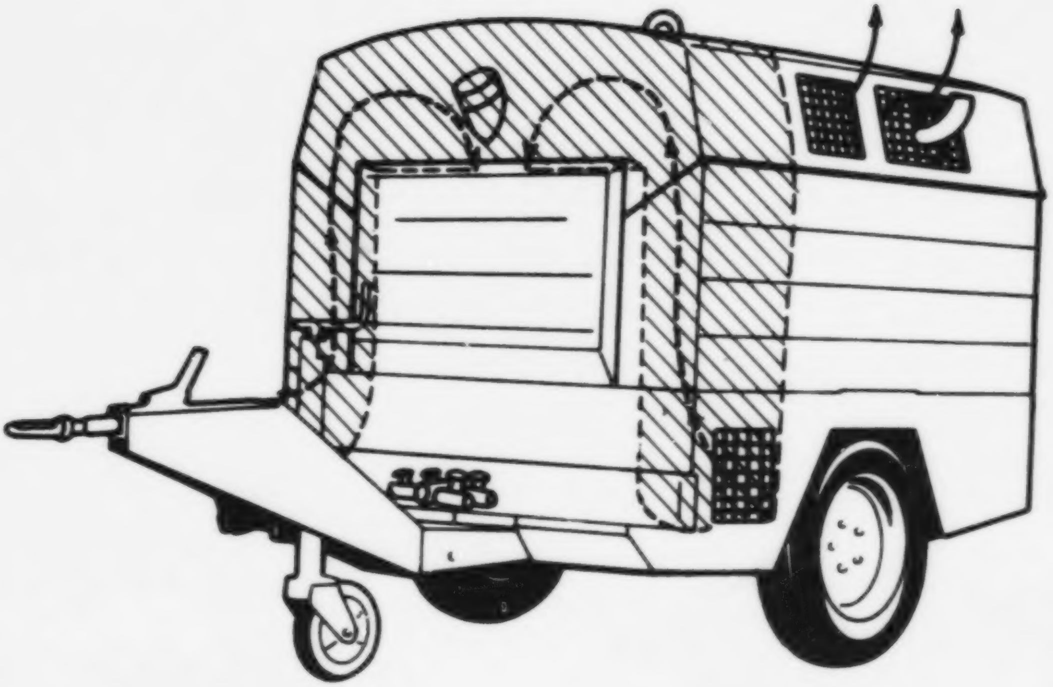


FIG. 31. COMPRESSOR INTAKE AND COOLING AIR DUCTING.

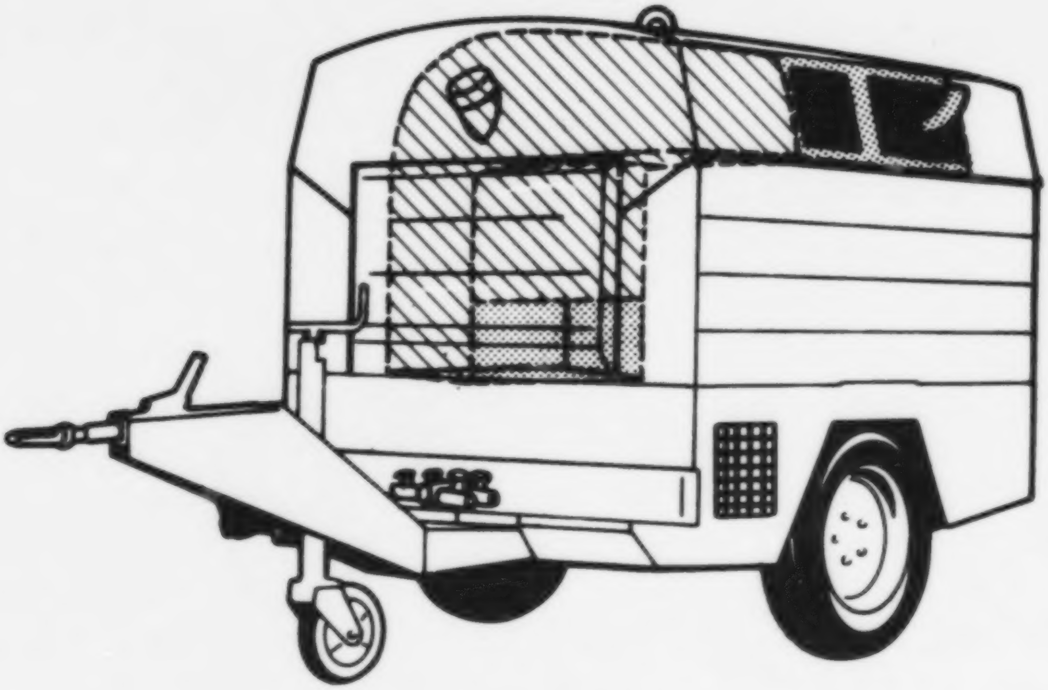


FIG. 32. ENGINE AND COMPRESSOR COOLING AIR EXHAUST DUCTS.

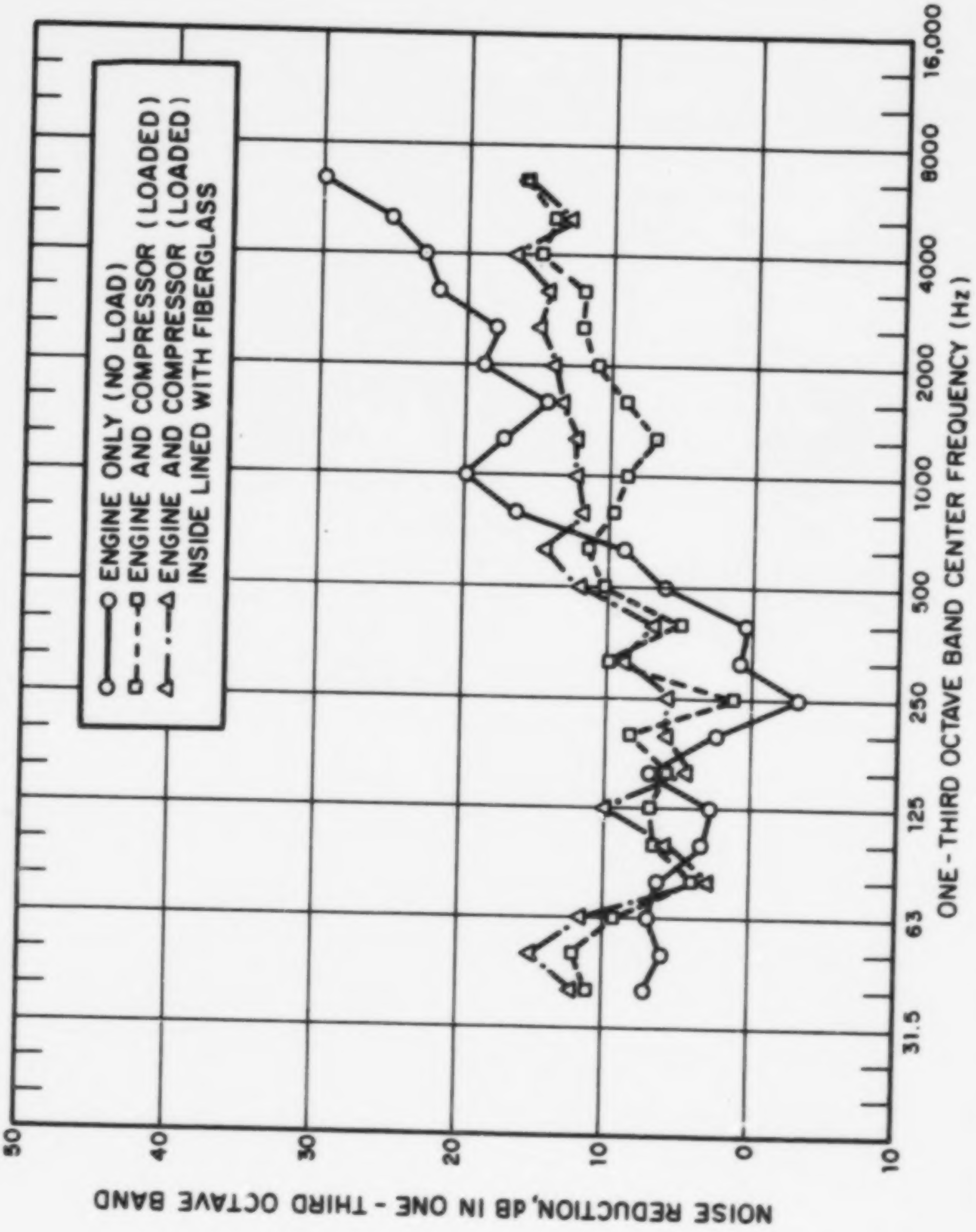


FIG. 33. NOISE REDUCTION 7 m FROM RIGHT SIDE WHEN ACCESS DOORS ARE CLOSED.

3.5 Possible Additional Noise Control Techniques

In our opinion, the noise control techniques applied to the Super Silensair unit are acoustically effective because all of the major sources have been treated to equivalent levels; that is, they all contribute nearly equally to the measured A-weighted sound pressure level. A 5- to 7-dB reduction in radiated sound could probably be obtained with the following noise reduction techniques:

1. Increase the damping of enclosure panels to reduce their resonant vibration levels and also to add mass, improving panel transmission loss and shifting resonant response to lower frequencies.

2. Increase the internal sound absorption by (a) treating a larger amount of the internal surface, and (b) using a thicker absorptive material. Also, cover absorptive material with a free-floating cover to prevent contamination (Bender, Patterson, and Kaye, 1974).

3. Use improved engine mounts, such as a 2-stage design (Bender *et al.*, 1974), to decouple the engine and compressor from the chassis.

4. Use a more effective exhaust muffler, such as a manifold muffler.

Appendices A and B discuss the costs associated with quieting the Atlas Copco Silensair compressor.

3.6 Operational Diagnosis

3.6.1 Cooling system performance

We were unable to determine directly the effects of the noise control treatments on the cooling system performance of this unit. Ducting of the cooling air appeared to be well designed. Flow losses seemed small compared to the pressure loss caused by forcing the cooling air over the cooling fins on the engine cylinders. At no time during the inside testing did the temperature warning light come on, even when the room temperature approached 100° F. A discussion with Atlas Copco representatives indicated that the maximum permissible ambient temperature for continuous operation is 105° F.

3.6.2 Fuel consumption

Tests performed at BBN gave a fuel consumption of 2.075 ± .02 gal/hr for the compressor operated at rated output. The manufacturer's representative estimated that there was, at most, a 2% fuel-consumption penalty associated with the noise control treatment. This unit is more efficient in converting fuel energy to compressed air when compared to the other units tested. Appendix C gives the comparison of operational performance.

3.6.3 Physical integrity of the enclosure

In general, the enclosure was well designed. Continuous hinges were used on all folding side panels and doors. Gasketing was made from a solid rubber extrusion rather than a foam strip, and appeared to be well applied and durable. Closing latches were effective and appeared to be durable.

The acoustic foam, however, was not sealed in plastic to protect it from contamination with oil and water. Such contamination would result in a decrease in performance of the absorptive lining.

The unit we tested was not equipped with a restraint to limit the degree to which the rear door could be opened. When the door was opened too widely, the door-hinge attachment flange bent. Had this bending been more severe, the gasket seal would not have been effective. The manufacturer's representative told us that newer models are equipped with a restraint system.

3.6.4 Effects of additional noise control on fuel consumption and cooling

We conclude that the four noise-reduction techniques proposed in Sec. 3.5 would have no significant effect on cooling system performance or fuel consumption. Damping of enclosure panels would not significantly change the flow area of the air ducts and so would not affect cooling capacity or fuel consumption. The mounts would have no detrimental effects. Sound absorptive material can be placed out of the path of major air flows to minimize its effect on cooling system performance. A more effective muffler might result in a slight increase in fuel consumption. Previous work by Donaldson (1968) and Holler (1970) indicates that increasing backpressure on the exhaust for naturally aspirated diesel engines can increase fuel consumption to 2%, at most, for the engines studied.

4. DIESEL ENGINE-POWERED AIR COMPRESSOR: >500 cfm

4.1 Introduction

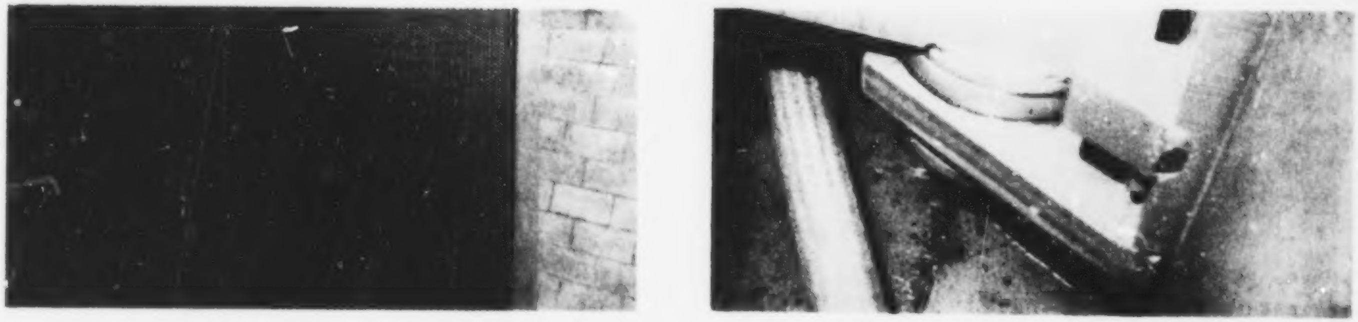
The "Blue Brute" 750-QTEX single-stage, portable, rotary-screw compressor is manufactured by Worthington CEI, one of the leading compressor manufacturers. Its GM Detroit Diesel Engine Model 6V-71 runs at 2100 rpm when the unit delivers its rated capacity of 750 cfm at 100 psig. The 750-QTEX is a quieted unit; it has been silenced to produce 75 dBA at 7 m (per ISO-2151-1972). Figure 34 is a photograph of the unit.

Among quieted diesel-powered compressors delivering greater than 500 cfm, the 750-QTEX is one of the quietest. It is only 1.5-dB noisier than the mean for the lowest decile, although its current list price is 3.5% above the mean price in the decile (Freeze *et al.*, 1974).

The technology by which the 750-QTEX has been quieted is also characteristic of the quietest compressors in its category. It has rubber engine mounts, nonrigid hose coupling, sealed doors, damped panels, interior sound absorption, silenced fan louvers for cooling-air intake and exhaust, 2-stage custom-designed muffler, bottom pan, and a special cooling fan. Some of these features are pointed out in the interior view photographs of Fig. 35.

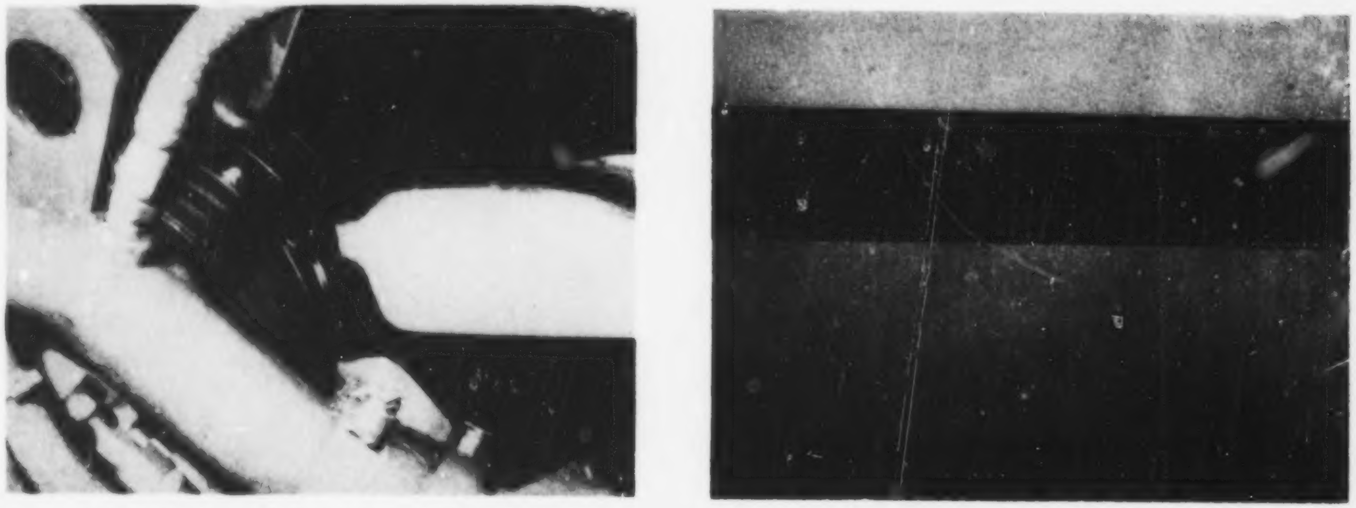
Consumption of 10 gal/hr of diesel fuel when operated at full load is characteristic of the GM 6V-71 running at its rated hp.

FIG. 34. WORTHINGTON "BLUE BRUTE" MODEL 750-QTEX, A QUIETED DIESEL ENGINE POWERED COMPRESSOR DELIVERING 750 cfm.



(a) ACOUSTICALLY TREATED DOOR PANELS AND DOOR GASKET SEAL

(b) VIBRATION ISOLATING MOUNTS FOR DIESEL ENGINE



(c) TWISTED-BLADE PROPELLER-STYLE FAN

(d) ACOUSTICALLY TREATED COOLING AIR EXIT VENT

FIG. 35. QUIETING MECHANISMS USED ON WORTHINGTON MODEL 750-QTEX.

Air-to-boil measurements yielded a value of 122° F, indicating no excessive penalty on operating temperature due to increased noise control. This is probably due to the special fan and other cooling-system improvements, which are more efficient than those originally provided with the engine.

Additional length is required to accomodate the silenced cooling-air intake louvers, and a nominal weight penalty is associated with the special muffler and additional sheet metal.

Our evaluation of the 750-QTEX is composed of (a) measurement of general characteristics with respect to noise, and operational and physical features that might have been influenced by the quieting design; (b) determination, by noise diagnosis, of the effectiveness of the present quieting and of the possibility of further quieting; (c) estimation of costs involved in present quieting and costs to obtain further quieting.

4.2 Present Noise Characteristics

Figure 36 presents data obtained by a modified form of procedure ISO-2151-1972, described in Appendix D, Report No. 2795a. According to these data, the 750-QTEX produces an average of 73.2 dBA at 7 m. Important features of these data are that the overall sound radiation is not very directive around and above the machine, and that its average sound level is 1.8-dBA less than the manufacturer's specification of its noisiness.

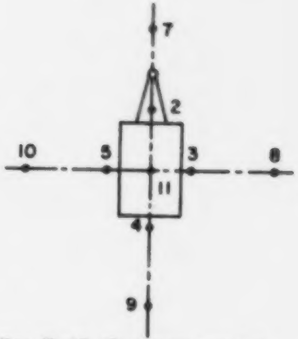
A-weighted radiated sound power of the 750-QTEX is 100.2 dB re 10^{-12} watts, as measured by the procedure described in

ONE-METER LEVELS

Level	Pos.	2	3	4	5
Idle	dBA	76	80	74	79
	Linear	86.5	89	86	87
Full Power	dBA	83.5	86	81.5	85
	Linear	89.5	94	94	92

SEVEN-METER LEVELS

Level	Pos.	7	8	9	10	11	Bkgd.
Idle	dBA	65.5	69.5	66	68	66	58.5
	Linear	80	84	82	82	82	69
Full Power	dBA	74.5	74	71.5	72.5	72.5	
	Linear	84	83	83	81	82	
Full Power	O.B.Ctr Freq(Hz)						
	32	66	65	67	67	63	60
	63	73	70	73.5	69.5	71	65
	125	82.5	78.5	81.5	76.5	75.5	64
	250	72.5	78	72.5	75.5	75.5	61.5
	500	69.5	72	70.5	71.5	71	56
	1000	62	69	63	67	66	52.5
	2000	63.5	60	58.5	62	62.5	45
	4000	68.5	59.5	58.5	58	59	40
	8000	68	54	54.5	53.5	55	27.5



Pos. 7-10: 7m from surface, 1.2m above plane

Pos. 11: 7m above surface

Pos. 2-5: 1m from surface, 1.5m above plane

FIG. 36. SUMMARY DATA ON NOISE EMISSIONS OF WORTHINGTON "BLUE BRUTE" MODEL 750-QTEX.

Sec. 2.1, Report No. 2795a. In this instance, we evaluated the ten sound levels on a 10-m hemisphere. Very little directivity is evident for overall levels; sound pressure levels measured at the ten points have a deviation of less than 0.7-dBA rms and a range of 2 dBA. If the radiated power is evaluated, using the modified CAGI-PNEUROP test discussed in Sec. 2, BBN Report No. 2795a, the five measurements at 7 m given in Fig. 37 yield an A-weighted sound power of 99.9 dB re 10^{-12}, only 0.3-dB less than the 10-point, 10-m measurement. Three representative octave spectra on the 10-m radius are presented in Fig. 38. The octave-band spectra are similar but exhibit some differences.

Another example of deviation in the radiated sound with direction is the variation with height of sound pressure levels measured at 7 m. The A-weighted levels vary about 1.5 dB over a 1.2-m change in height. Figure 39 displays octave band spectra typical of the measured variation measured at 7 m from the side of the compressor (Position 8), at heights of 0.6 m, 1.2 m, and 1.8 m. Nonuniformity of spectral acoustic radiation is also evidenced in the data of Fig. 37, in which the linear and A-weighted 1/3-octave band spectra for five measurement positions are displayed.

In line with the previous results, we would expect that extrapolation of sound pressure levels to greater distances cannot be done simply on the basis of 6 dB per doubling of distance. Variations with height displayed in Fig. 39 and variations with position on a 10-m hemisphere displayed in Fig. 38 are indicative of ground reflection contributions that cause deviations from expected freefield decrement vs distance. Figure 40 illustrates three 1/3-octave spectra taken along a 270° radius at 4,

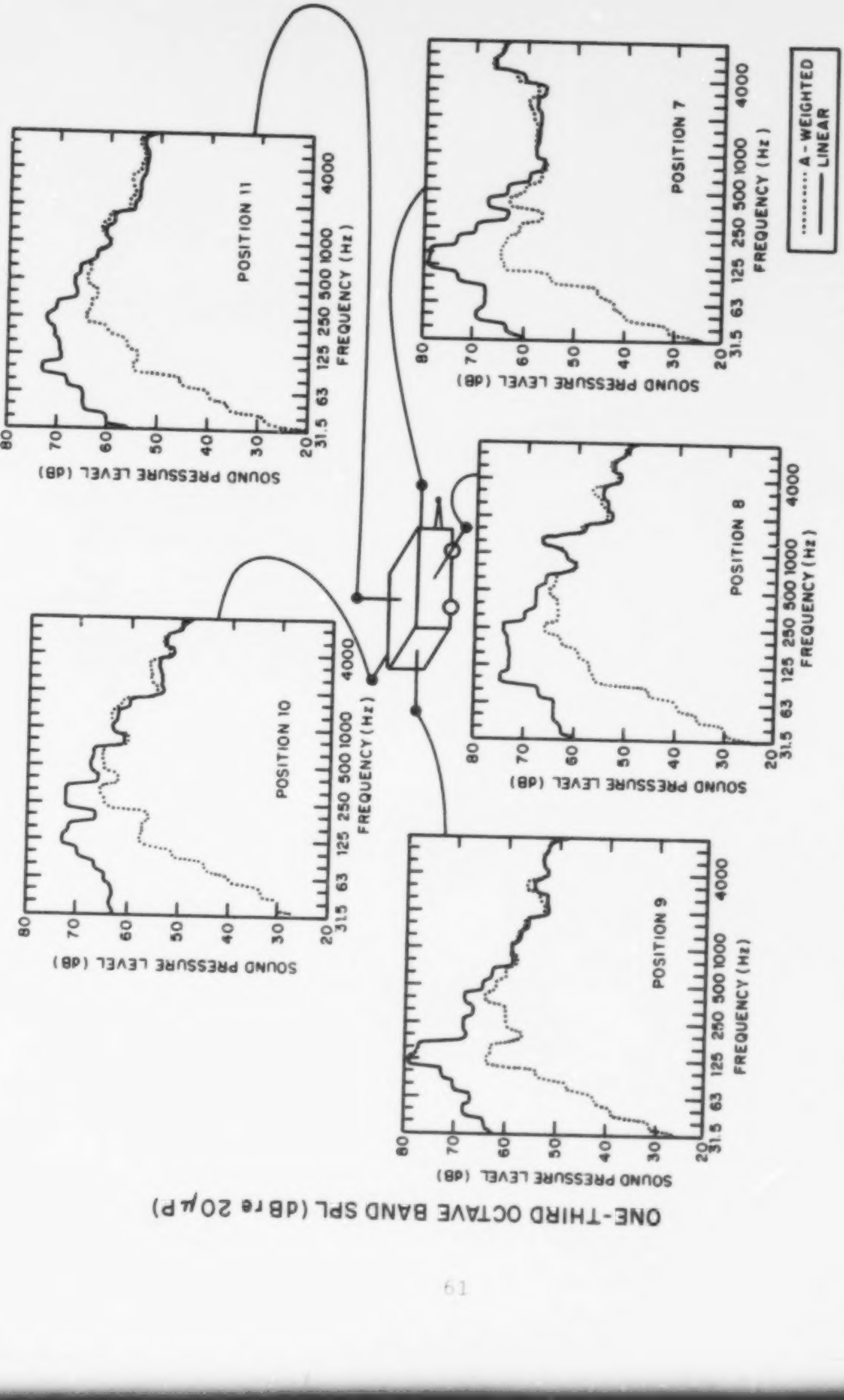


FIG. 37. LINEAR AND A-WEIGHTED 1/3-OCTAVE BAND NOISE SPECTRA AT FOUR CARDINAL POINTS AND OVERHEAD AT A DISTANCE OF 7 m FROM THE WORTHINGTON "BLUE BRUTE" MODEL 750-QTEX.

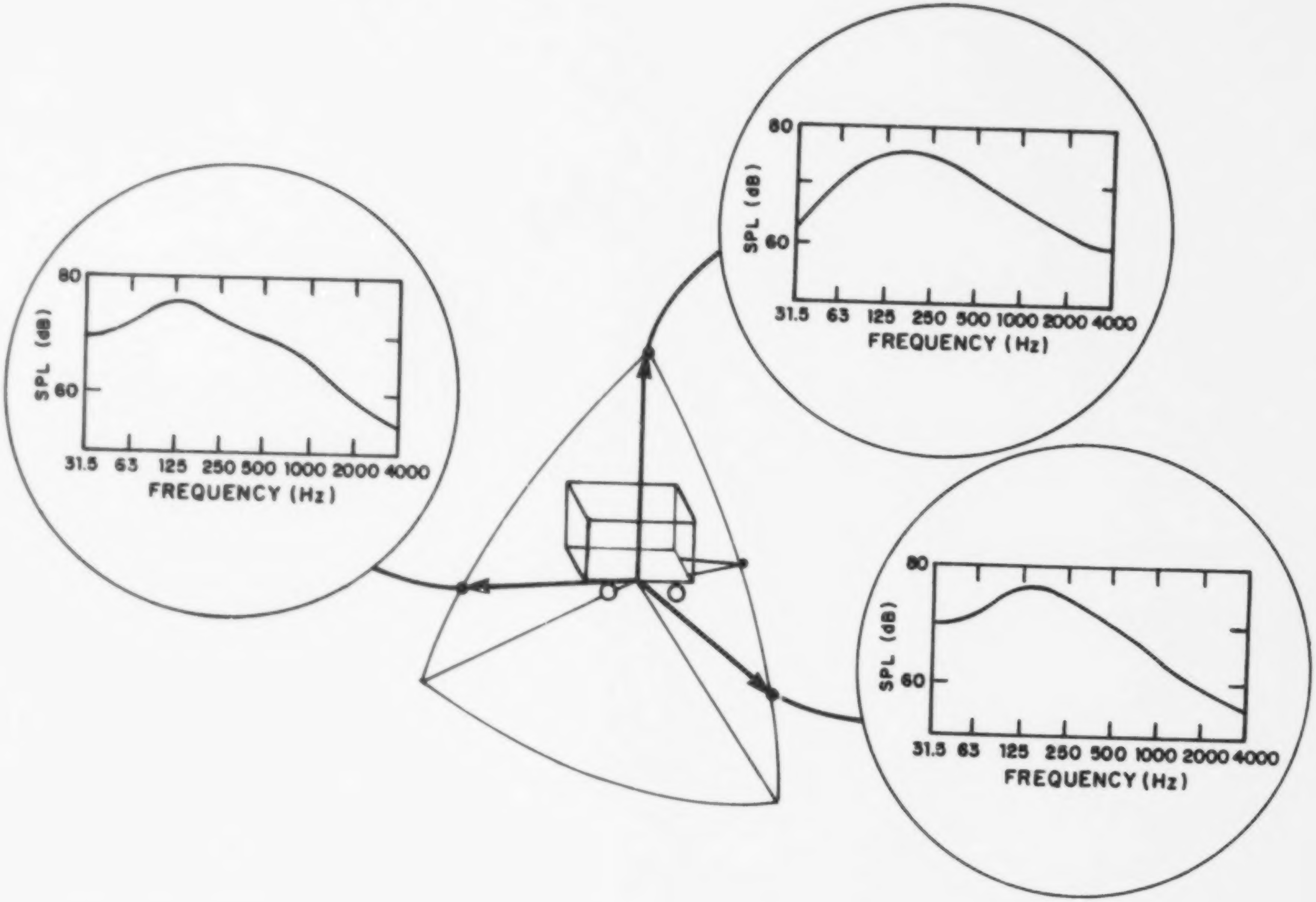


FIG. 38. OCTAVE BAND SPECTRA AT THREE POINTS ON A 10-METER HEMISPHERE SHOWING GENERAL UNIFORMITY OF RADIATION FROM WORTHINGTON "BLUE BRUTE" MODEL 750-QTEX.

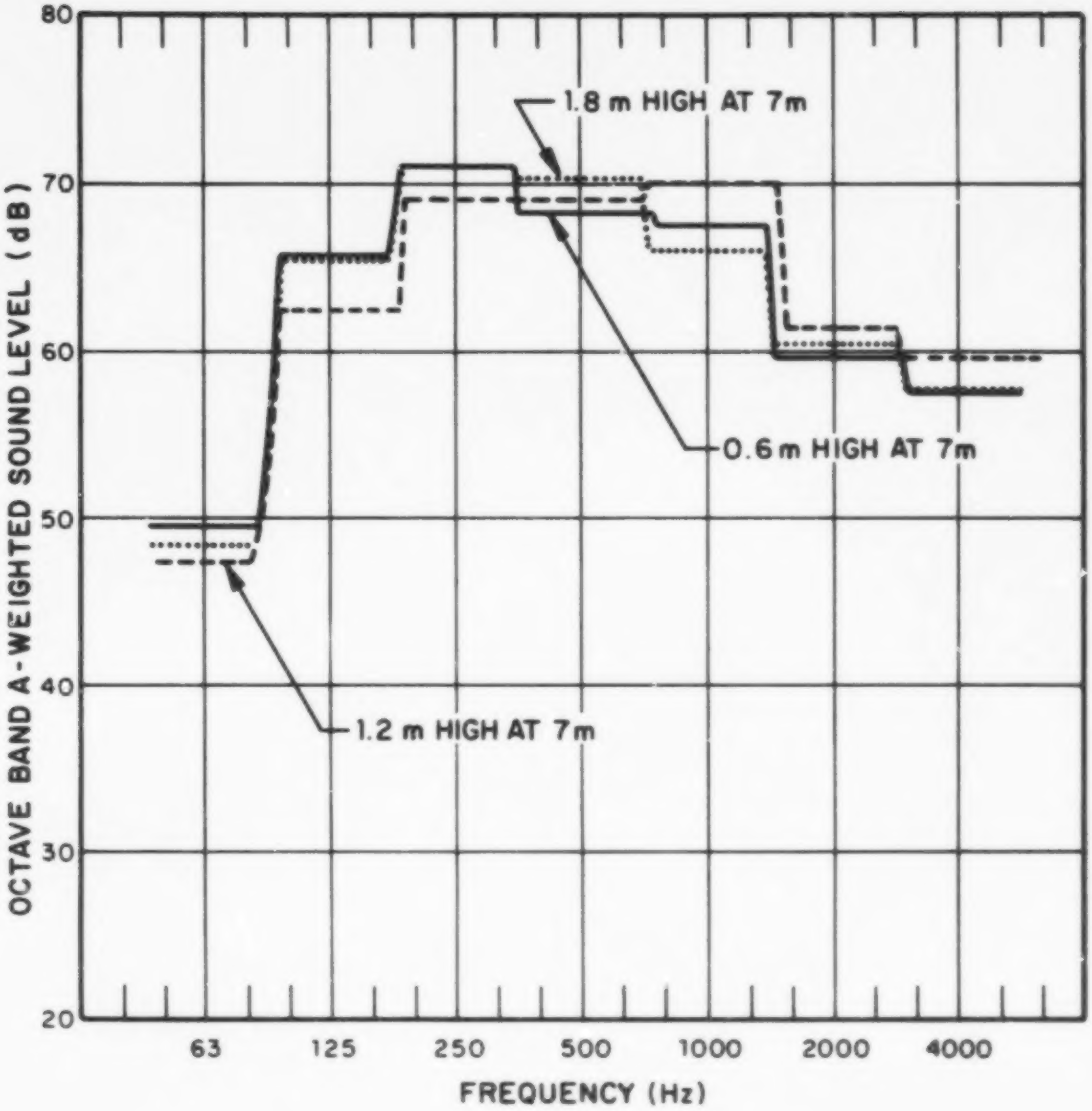


FIG. 39. A-WEIGHTED OCTAVE BAND SPECTRA AT THREE HEIGHTS SHOWING TYPICAL VARIATION WITH HEIGHT AT 7 m FROM SIDE OF WORTHINGTON "BLUE BRUTE" MODEL 750-QTEX.

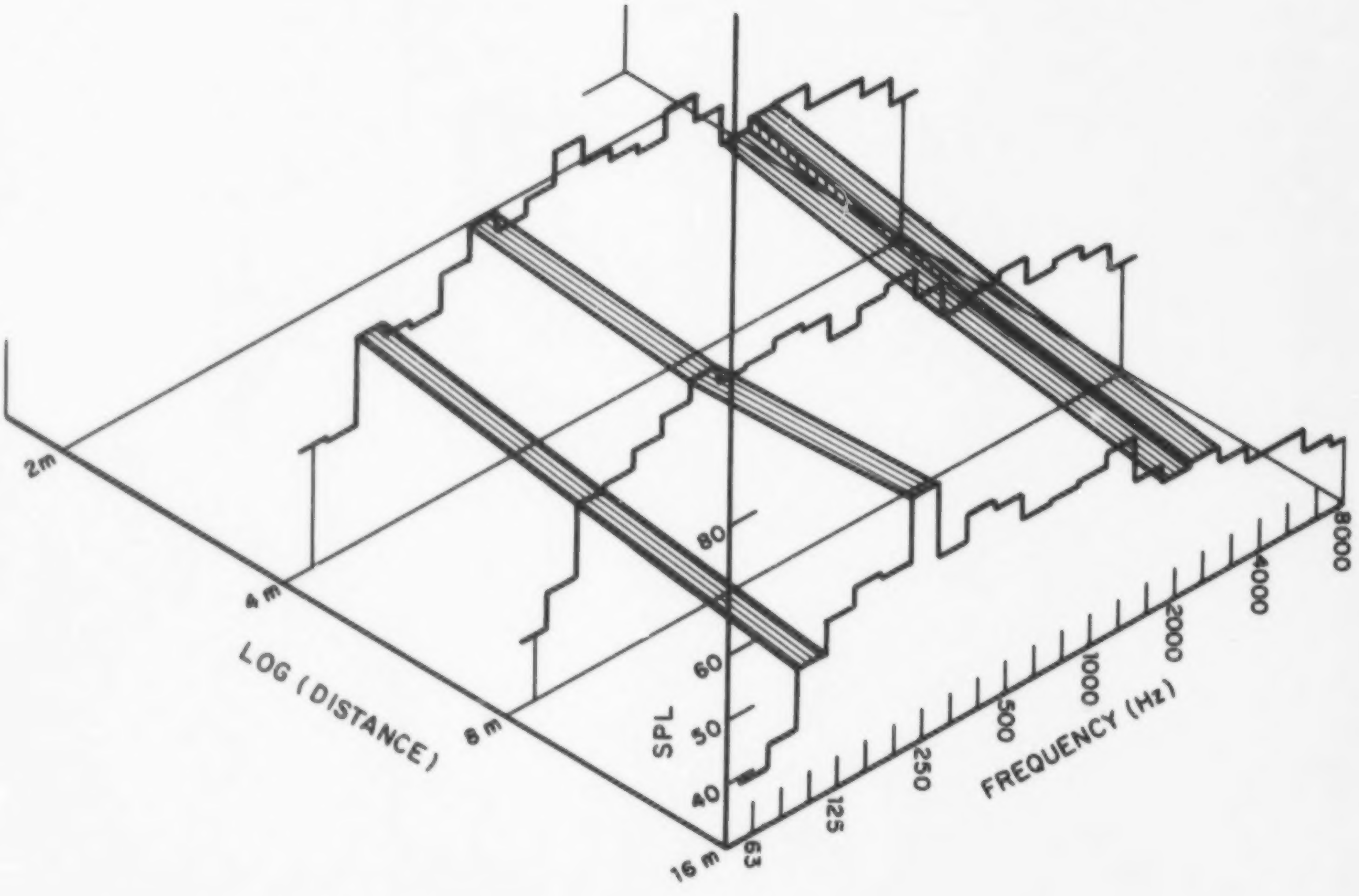


FIG. 40. A-WEIGHTED 1/3-OCTAVE BAND SPECTRA OF THE SOUND PRESSURE LEVEL AT 4, 8, AND 16 m ALONG A 270° RADIUS FROM THE CENTER OF THE WORTHINGTON 750-QTEX.

8, and 16 m from the center of the 750-QTEX. The A-weighted overall attenuation is 4 dB per doubling of distance, due primarily to anomalous accentuation in certain bands (e.g., 250 Hz, 500 Hz, etc.). Further discussion of the reasons for this deviation in sound level decrease with distance is presented in Report No. 2795a.

This evaluation of the Worthington 750-QTEX can be used to determine *in situ* sound levels for maximum air delivery only if the machine is operated with the doors clasped shut. Idle operation with doors shut produces A-weighted sound pressure levels about 5-dB lower; sound levels that occur when the doors are left open average 10- to 12-dB higher. The three 1/3-octave band spectra of Fig. 41 illustrate the marked effect of operating conditions on noise emissions. They were measured 1.2 m above the ground at 7 m from the right-hand side panels.

4.3 Identifying the Relative Contribution of the Major Sources of Noise

Remedial measures for noise reduction, especially noise source removal, can only be investigated when the separate contributions of the major noise-producing elements of the compressor assembly have been identified. When there is a characteristic frequency spectrum from each major distinguishable source, a narrowband analysis is useful; one attempts to recognize the spectral signature of each source in the narrowband spectrum of the farfield measurement.

The Worthington 750-QTEX presents a formidable problem in this respect. Each source – engine, fan, rotary-screw compressor, etc. – generates a line spectrum which contains almost the same

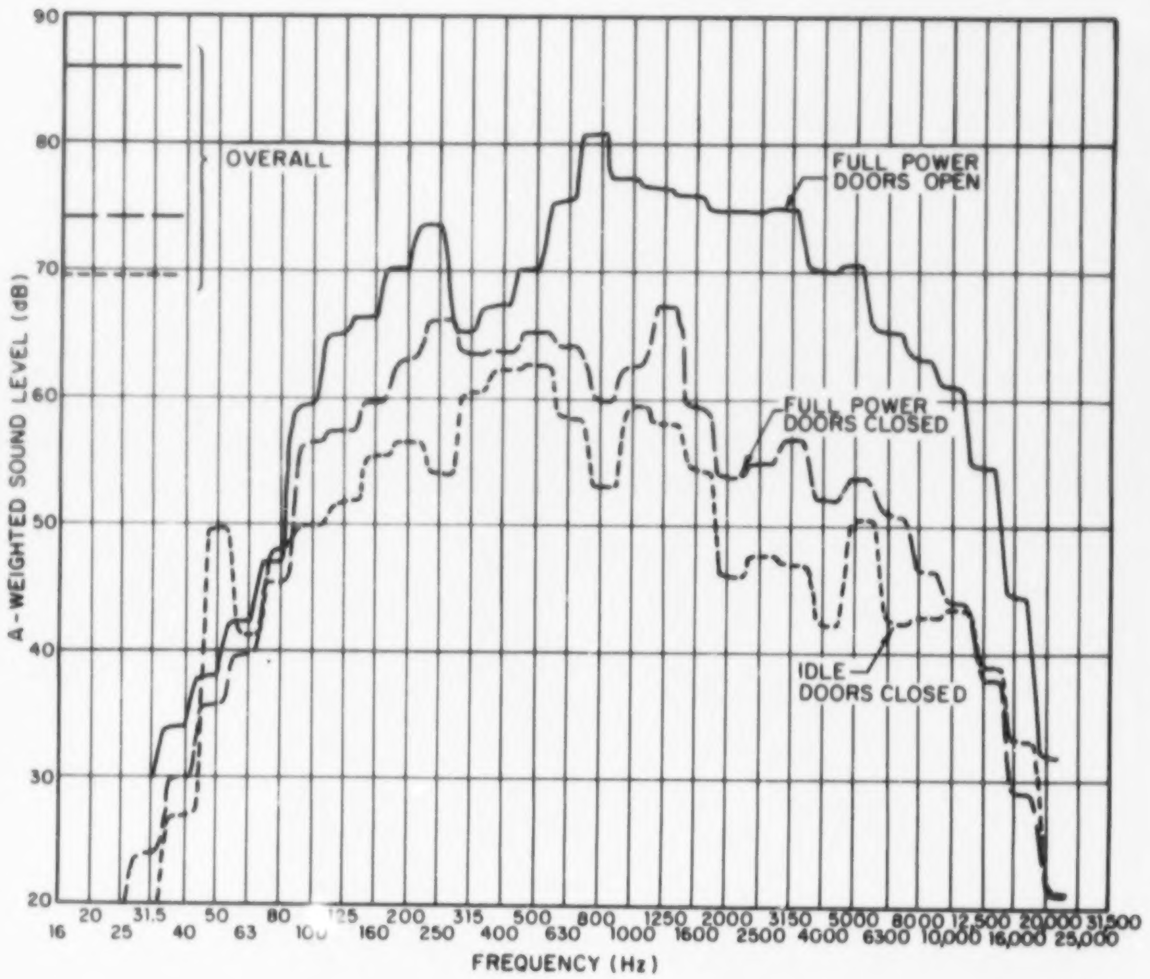


FIG. 41. A-WEIGHTED 1/3-OCTAVE BAND AND OVERALL NOISE LEVELS FROM THE WORTHINGTON 750-QTEX FOR THREE OPERATING CONDITIONS: IDLE AND FULL RATED DELIVERY WITH DOORS CLOSED, AND FULL RATED DELIVERY WITH DOORS OPEN.

spectral lines. Passage rates for the fan blades and for the main rotor of the rotary-screw compressor are 140 and 144 Hz, respectively. Engine firing rate is 210 Hz and supercharger blower passage rate is 215 Hz. Furthermore, since nonlinearities cause consideration modulation, mostly by the main shaft at 35 Hz, there are overlapping sidebands throughout the spectra. In addition, because the unit has been systematically quieted, we would expect the relative contributions to be approximately equal. Examples of narrowband spectra presented in Fig. 42 indicate the complexity. Each spectrum is presented over a dimensionless frequency parameter that expresses the frequency as a multiple of the repetition rate of the relevant fundamental noise-producing mechanism. Marked along the top of each spectrum are the principal harmonics and side bands associated with main-shaft rotation rate.

Because of the difficulty in obtaining reliable results by comparing the spectra of Fig. 42 with farfield spectra, we operated the fan separately and adapted correlation techniques used in other work (Scharton, 1973).

4.3.1 Fan noise

To cool the engine and rotary-screw compressor on the Worthington 750-QTEX, a 122-cm diameter, 6-bladed, propeller-type fan rotating at 1400 rpm (full load) draws air through a radiator core from an intake plenum that is opened to the outside through silenced louvers. The air exits through silenced louvers in the side of the front plenum chamber and through a vent on top of the plenum. The top vent, also silenced, opens pneumatically when the compressor is operating.

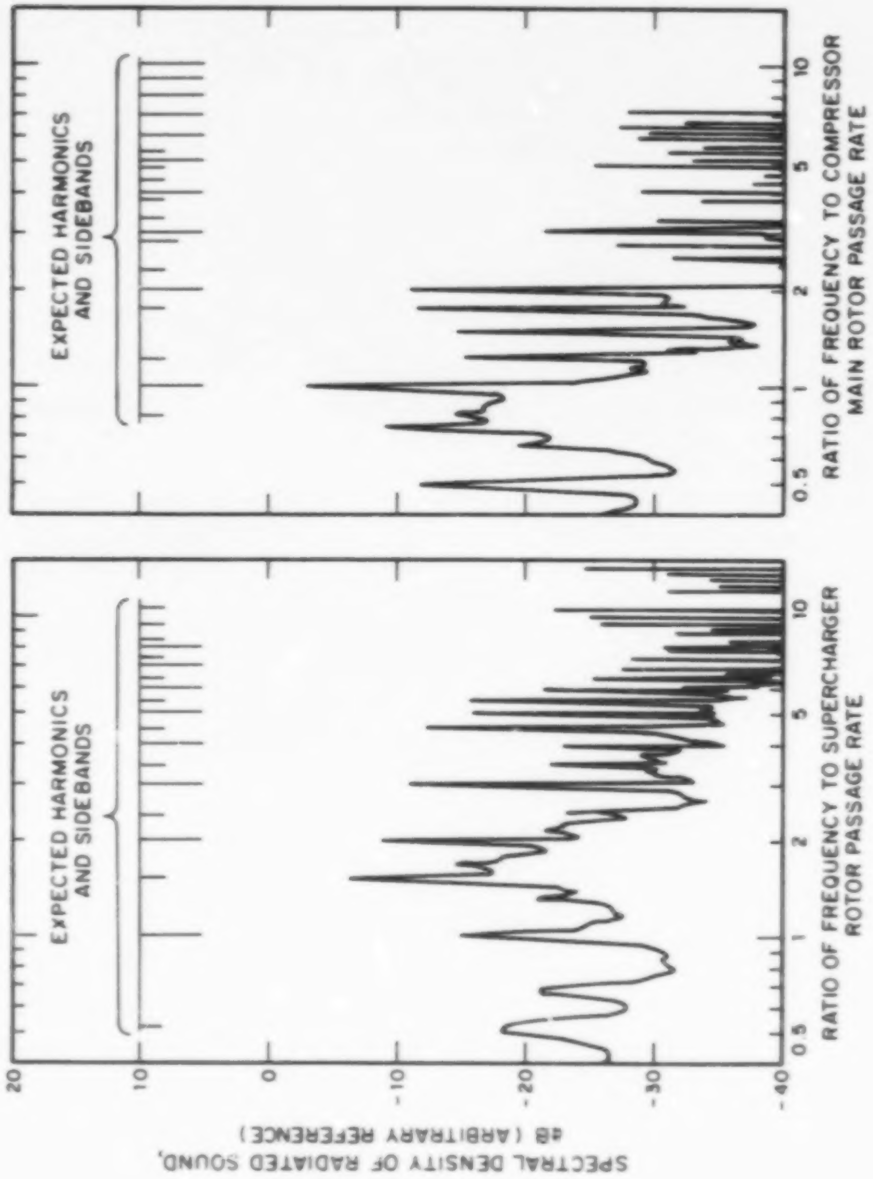


FIG. 42. NARROWBAND SPECTRA OF TWO SOURCES OF THE NOISE EMISSION OF THE WORTHINGTON 750-QTEX.

The original fan used with the GM Detroit Diesel 6V-71 is replaced by the fan pictured in Fig. 36c. The larger diameter and twisted blades of the new fan, and the design of the fan shroud, radiator, and silenced cooling-air intake plenum constitute appropriate elements of an efficient air pumping system. The improved pumping efficiency allows the fan to operate with reduced tip speed.

To evaluate the fan noise source level, we used an electric motor to drive the fan without operating any other noise-generating mechanisms of the compressor. This technique is discussed in Sec. 2.5. In the case of the Worthington 750-QTEX, we found no difficulty in replicating normal operating configuration while driving the fan alone. The fan pulley is on an idler shaft and drives no water pump or other mechanisms, and the fan shroud can be left intact. It was necessary to scale the data using 10 log V^6 sound level to fan speed relation to compensate for the 83% speed limitation of the electric motor drive system. Operation of the cooling fan, at full speed, requires approximately 11 hp.

Comparisons of noise in octave bands from the compressor operating at full power vs noise from the fan alone summarize our findings. Figure 43 shows that cooling-fan noise remains a major source of noise radiating in the direction of the fan axis. It is a minor but significant contribution to the noise radiating to the sides of the compressor (90° and 270°).

4.3.2 Other noise sources

To apply correlation techniques for determining component source levels requires, first, that we restrict consideration

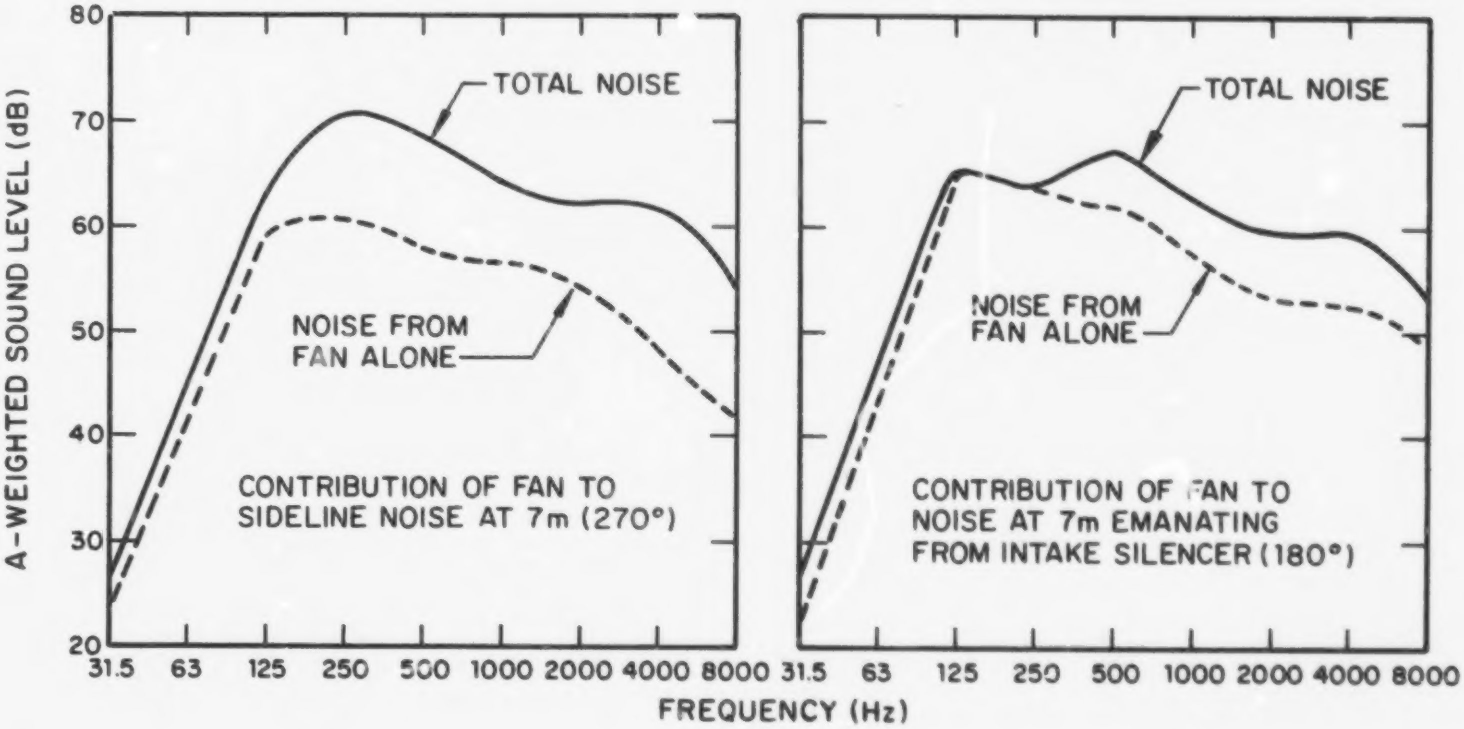


FIG. 43. COMPARISON OF TWO POSITIONS OF A-WEIGHTED OCTAVE BAND SPECTRA OF FULL POWER NOISE AND NOISE GENERATED BY THE FAN ALONE.

to the prime noise sources; then, by means of extensive measurement and calculation, we obtain correlations between farfield sound pressure and locally radiated sound off each prime source. We then use these results to determine the relative contribution of each source to the sound power in the far field.

Distinguishable assembles that are likely to be substantial contributors to the farfield noise are fan, muffler, engine, and rotary-screw compressor. A most important mechanism of acoustic radiation, shared by the muffler, engine, and compressor, is case radiation. This is the dominant mechanism for radiation from the present muffler; the large surface area is acoustically efficient, and exhaust noise from pressure fluctuations at the stack is well attenuated. For the engine and compressor, case-radiated noise dominates intake noise, which is well attenuated in two air-intake filters. Hence, the prime sources we must add to the independent estimate of fan noise are muffler-case radiation, rotary-screw compressor-case radiation, and engine-case radiation.

Conceptually, the correlation technique uses the locally obtained acoustic signature of each source as a kind of mathematical filter, by means of which the contribution of each source type is evaluated. Although extensive correlation measurement between candidate sources can facilitate distinguishing them from one another, a less tedious approach is to isolate the characteristic sound of each candidate source. In the present study, we used the acoustic duct method of Thien (1968). In our adaptation, the sound off the case of each source is detected in a fiberglass-lined duct made of plywood and lead sheet, which attenuates the sound from other sources. In this way, we obtain from the locally radiated sound of the muffler shell, compressor housing, or engine block a source characteristic that can be correlated with the farfield sound.

For the engine-case radiation, we chose an area about midway along the engine axis at the water-plug line. To verify that this was representative of case radiation from the engine, we also correlated the far field with ducted noise from the supercharger and valve covers. Similarly, we compared correlation of the noise ducted from the compressor air-inlet region with that from the main housing. Because of the physical uniformity of the muffler, we made no differential measurements, we simply applied to sound duct to an area about two-thirds along the length of its first stage.

Reduction of the extensive correlation measurements yields A-weighted octave band spectra of the contributions to the farfield noise of muffler shell, engine, and rotary-screw compressor. These, together with the fan noise, are summarized in Fig. 44. Although these data are only for the 7-m position on the 90° side, they are adequate to assess the requirements of further noise reduction.

The contribution of present source levels to the overall measured sound level is given in Fig. 45. The predominant source of noise is the muffler shell; it is followed by the engine casing noise.

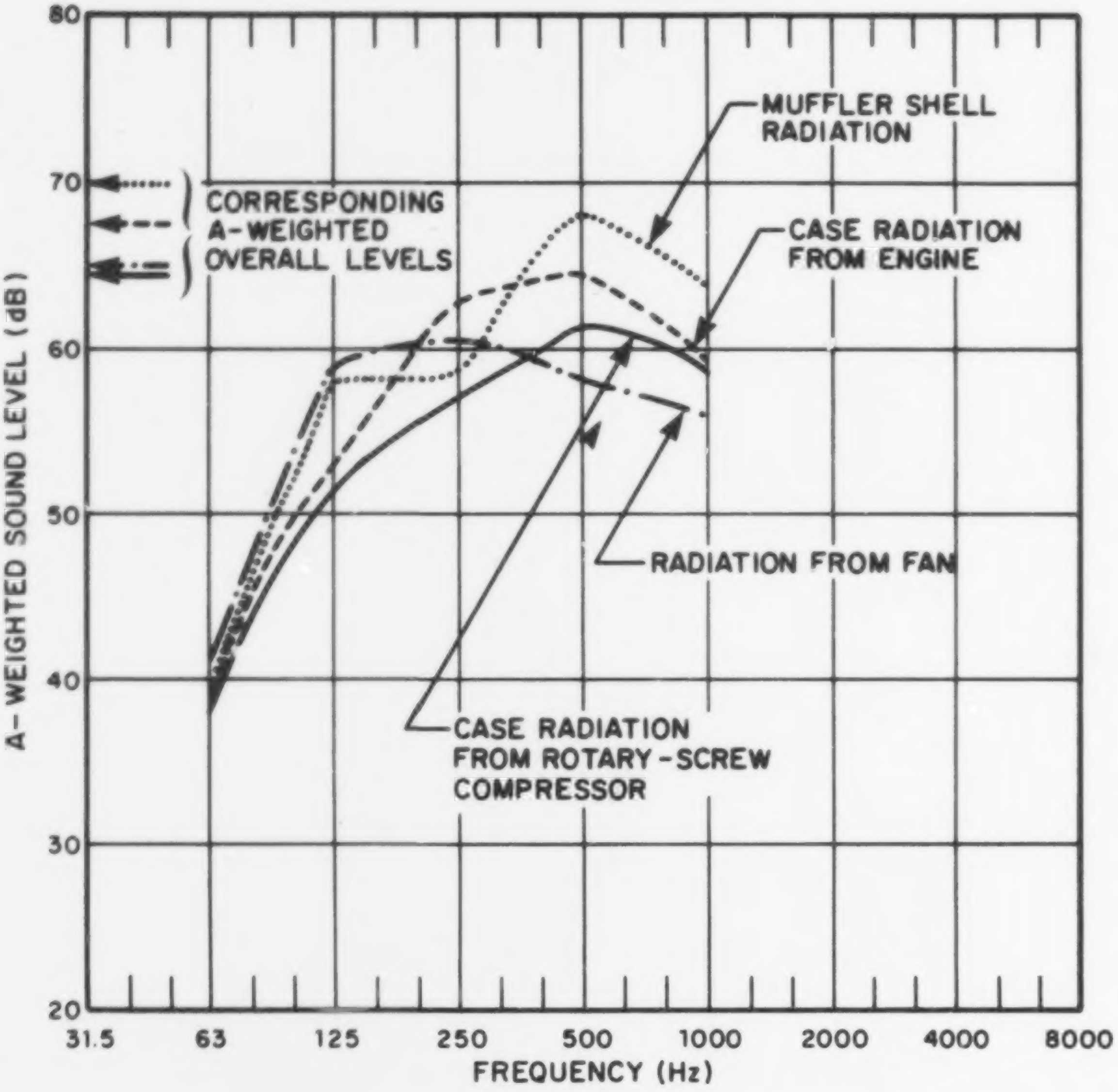


FIG. 44. A-WEIGHTED OVERALL LEVELS AND OCTAVE BAND SPECTRA SHOWING CONTRIBUTION TO NOISE AT 7 m (270°) FROM MAJOR SOURCES IN THE WORTHINGTON 750-QTEX.

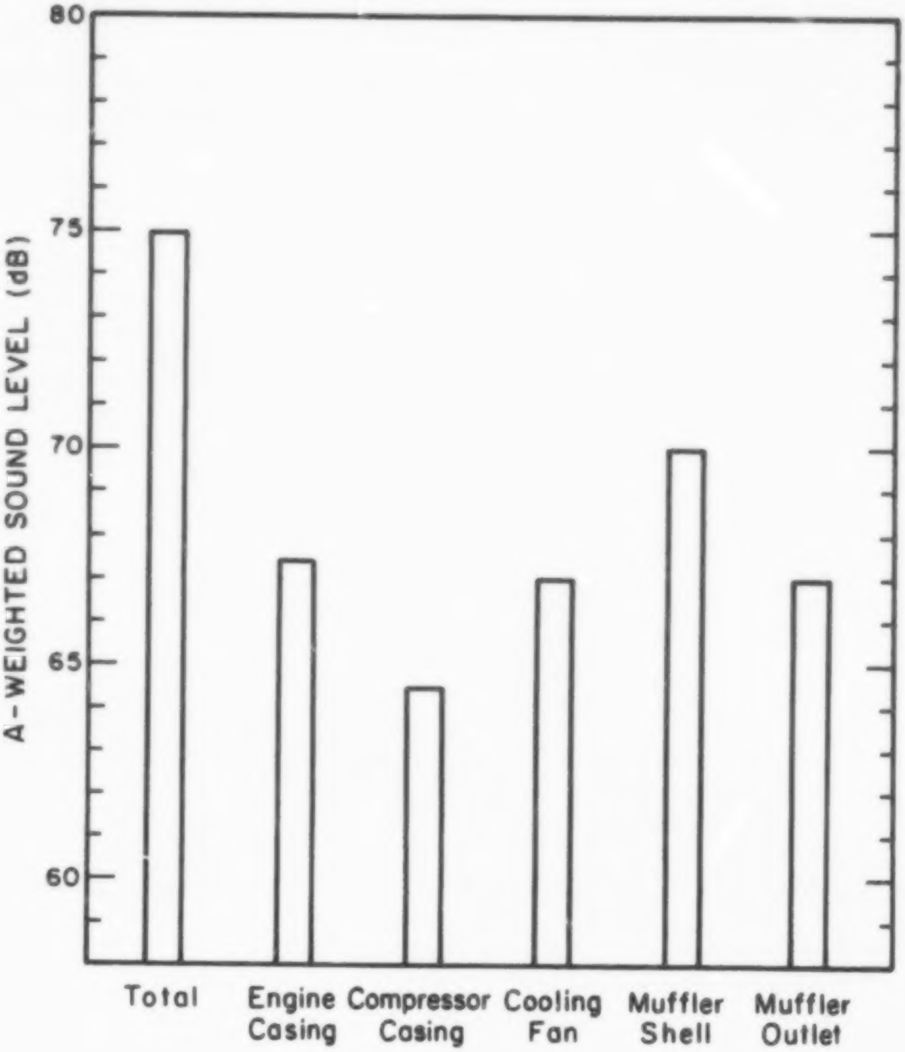


FIG. 45. BAR GRAPH OF 750-QTEX COMPONENT SOURCE LEVELS.

4.4 Evaluation of Present Quieting

Together, the special cowling, fan, and reducing pulley for the cooling-air intake constitute an application of the technology of noise-source reduction. The acoustic enclosure is an example of the technology of noise insulation. The muffler involves both technologies. We chose to examine in depth the enclosure and fan.

4.4.1 Acoustic enclosure

The primary quieting mechanism on this machine is the acoustic enclosure. Flexible couplings and shock mounts minimize structureborne noise, and a nonstandard cooling fan and cowling reduce fan noise; but all primary noise generators – engine, rotary-screw compressor, and fan – are enclosed in an approximately $3 \times 2 \times 2$ m^3 acoustic enclosure. This acoustic enclosure, with silenced louvers for the admission and exit of cooling air, is composed primarily of 16-gauge steel plate and a commercial acoustic material that increases the transmission loss of the steel plate and reduces the internal reverberant field by absorption. Indicative of the role played by this enclosure is the increase from 74 to 86 dBA when the doors are opened (see Fig. 41).

To evaluate the enclosure, we placed a loudspeaker inside to provide a known noise source. Levels inside, measured at two locations, suggest an interior reverberant field (in spite of the acoustic material placed inside the enclosure for absorption). Levels outside at 1 m and 7 m on all sides, when compared with interior levels, provide an evaluation of the noise reduction provided by the acoustically treated side panels and silenced louvers.

Figure 46 displays noise reduction vs frequency for end positions (0° and 180°) and side positions (90° and 270°). Two of the curves represent insertion loss of the sound measured at 7 m (1.2 m above the ground), i.e., the reduction relative to what would occur if the interior sound were not enclosed. The noise reduction curve is obtained as the difference between the interior sound pressure level and that measured 1 m from the enclosure door.

The measurements indicate that noise reduction referred to 7 m is essentially what would be expected from untreated 16-gauge steel (35 dB at 1000 Hz). Furthermore, the leakage provided by the exit louvers apparently permits transmission above 500 Hz. In this frequency range, noise measured at 7 m from the side (90° and 270°) is about 10-dB louder than should be expected, if the noise reduction of a full 16-gauge steel-panel enclosure were operative. Comparison of measured noise reduction through the side panel and that expected of 16-gauge steel above 1500 Hz also supports the hypothesis of flanking from the exit louvers. There is also absorptive material (foam) that could be reducing the interior reverberant field; it would contribute to the effectiveness of the enclosure, but not show up in measurements that compare interior with exterior level. However, interior absorption does not appear to be large. Measurements inside show a reverberant field, and the ratio of absorptive area to hard surface is small. Measurements of the acoustic characteristics of foams with covering films show that when the film is adhered to the foam, severe degradation of acoustic absorption results (Bender *et al.*, 1974).

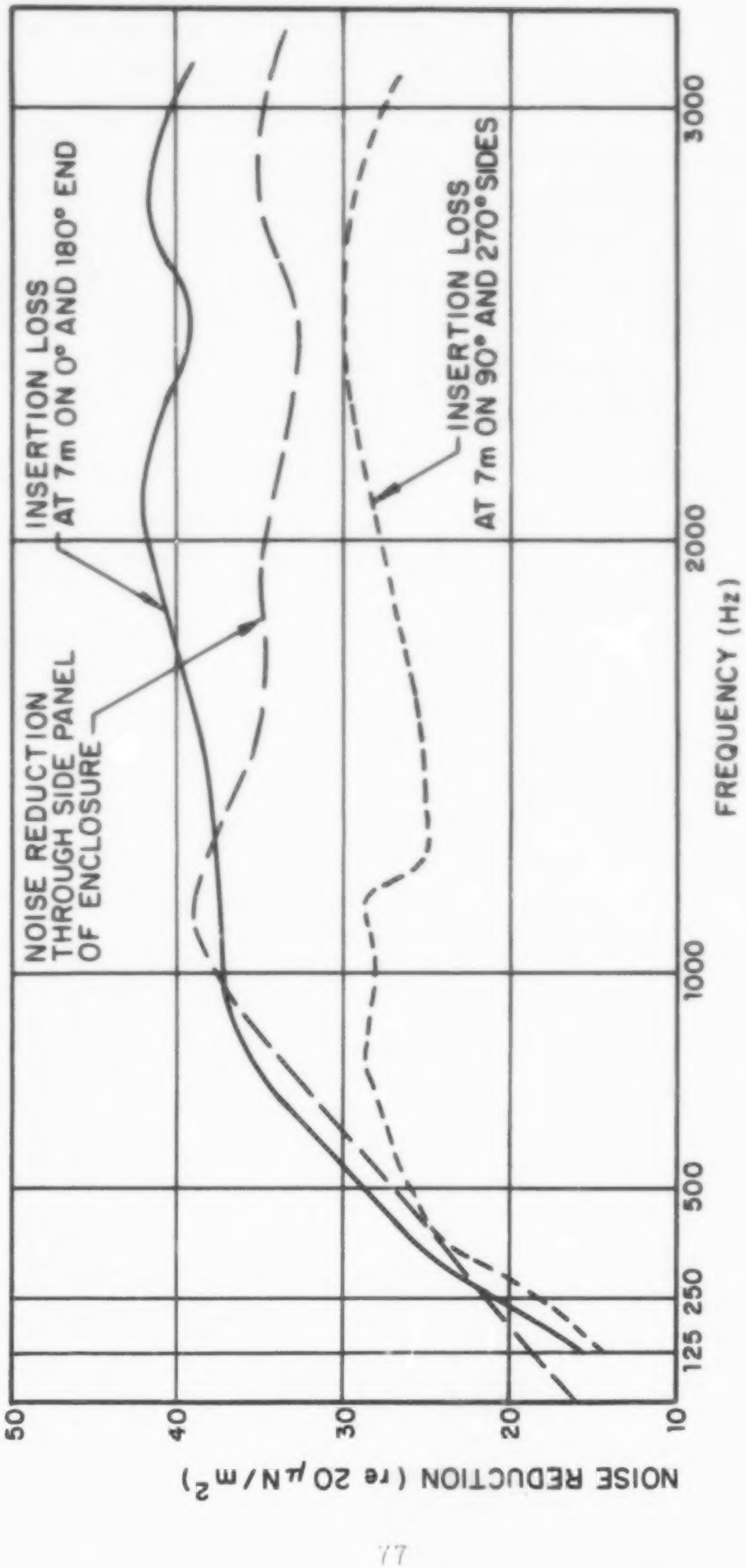


FIG. 46. NOISE REDUCTION DUE TO ACOUSTIC ENCLOSURE ON WORTHINGTON 750-QTEX, ATTENUATION AT 7 m, AND TRANSMISSION LOSS THROUGH SIDE PANEL.

There is another way in which the acoustic treatment of the enclosure may be effective, yet not show up in the above comparison. Some damping of mechanical vibration of side panels is probably provided by Blachford *Baryfol*, used for absorption and added transmission loss on the side panels. Without the *Baryfol*, there might have been more radiation. The combination of flexible couplings, shock mounts, and panel damping could be the most significant noise-reduction mechanism, in addition to the 16-gauge steel enclosure and the muffler.

Costs of the enclosure are summarized in Appendix A. They indicate that the noise reduction shown in Fig. 46 was obtained at a cost of $404 for materials and 100.8 hrs of labor to assemble beyond the standard enclosure.

4.5 Determination of a Strategy for Further Quieting

Further quieting should be possible by a more thorough application of technology that has already been applied and by additional design features. We examine these separately, considering improvements on the enclosure, fan, and mufflers.

4.5.1 Enclosure redesign

The enclosure presently provides adequate noise reduction of engine and compressor airborne sound, except at the cooling-air intake and exit ducts. Additional noise reduction is possible with design improvement of both the ducts and the material used for acoustic absorption.

The present cooling-air duct arrangement has splitters (acoustically absorptive panels inserted parallel to the flow) spaced for a duct height-to-length ratio of approximately 1/4. With optimal conditions, this arrangement would provide an insertion loss of about 12 dB (Beranek, 1971). Additional attenuation of internal airborne sound would be possible by decreasing the height-to-length ratio with additional splitters placed closer together. This design would also require a larger plenum opening if the splitters are the same thickness. In addition, the splitters could be constructed in an interlocking diamond pattern to block direct line-of-sight sound radiation.

Adding acoustic absorptive treatment to the enclosure interior will reduce both the reverberant buildup inside and subsequent exterior radiation, either through the enclosure walls or cooling-air ducts. As discussed previously, the present foam material used for acoustic absorption has a film adhered to the surface, which severely degrades the absorptive properties of the foam. By adding a spacer between the film and the foam surface to allow free movement of the film, the film becomes acoustically transparent to impinging sound waves for the frequencies of interest. The sound-absorbing qualities of the foam may then reduce the reverberant buildup of internal enclosure noise.

To accomplish these goals, additional material and labor will be required. Estimated material requirements are an additional quantity of sound absorbing foam to cover a total surface area of 32 m^2 ($565). For labor costs, we estimate additional labor equal to that used in the present assembly of the enclosure (100.8 hrs).

4.5.2 Performance of an improved fan design

Although the cooling fan and associated hardware on the Worthington 750-QTEX show evidence of design for noise reduction, considerable improvement should be achievable with more attention to the primary noise-producing mechanism: a fan's noise is dependent above all on the blade-tip tangential velocity.

Reduction of blade-tip velocity is practical if the cooling requirements can be met at lower blade-tip velocity.

To determine if much improvement is possible, we measured the loading on the present fan. This is given by the propeller-disc thrust coefficient,

$$C_T = \Delta P/\tfrac{1}{2}\rho v_t^2 \quad ,$$

in which ΔP is the pressure rise across the fan, ρ is the density of air, and v_t is the blade tangential speed (all measured in any set of consistent units). For the Worthington 750-QTEX, we obtain $C_T = .0611$, which indicates that the fan is lightly loaded. Our experience with quieting a diesel truck fan indicates that about three times this loading would be acceptable, although the fan might require more power to drive.

A fan of the same diameter, larger chord, and higher pitch ratio, running at about 800 rpm (at full load) would experience three times the loading. Such a fan design could result in a shift in the spectrum to lower frequencies and a reduction in noise output of a little more than 14 dB. In Fig. 47, we illustrate the octave band spectrum that could then be expected

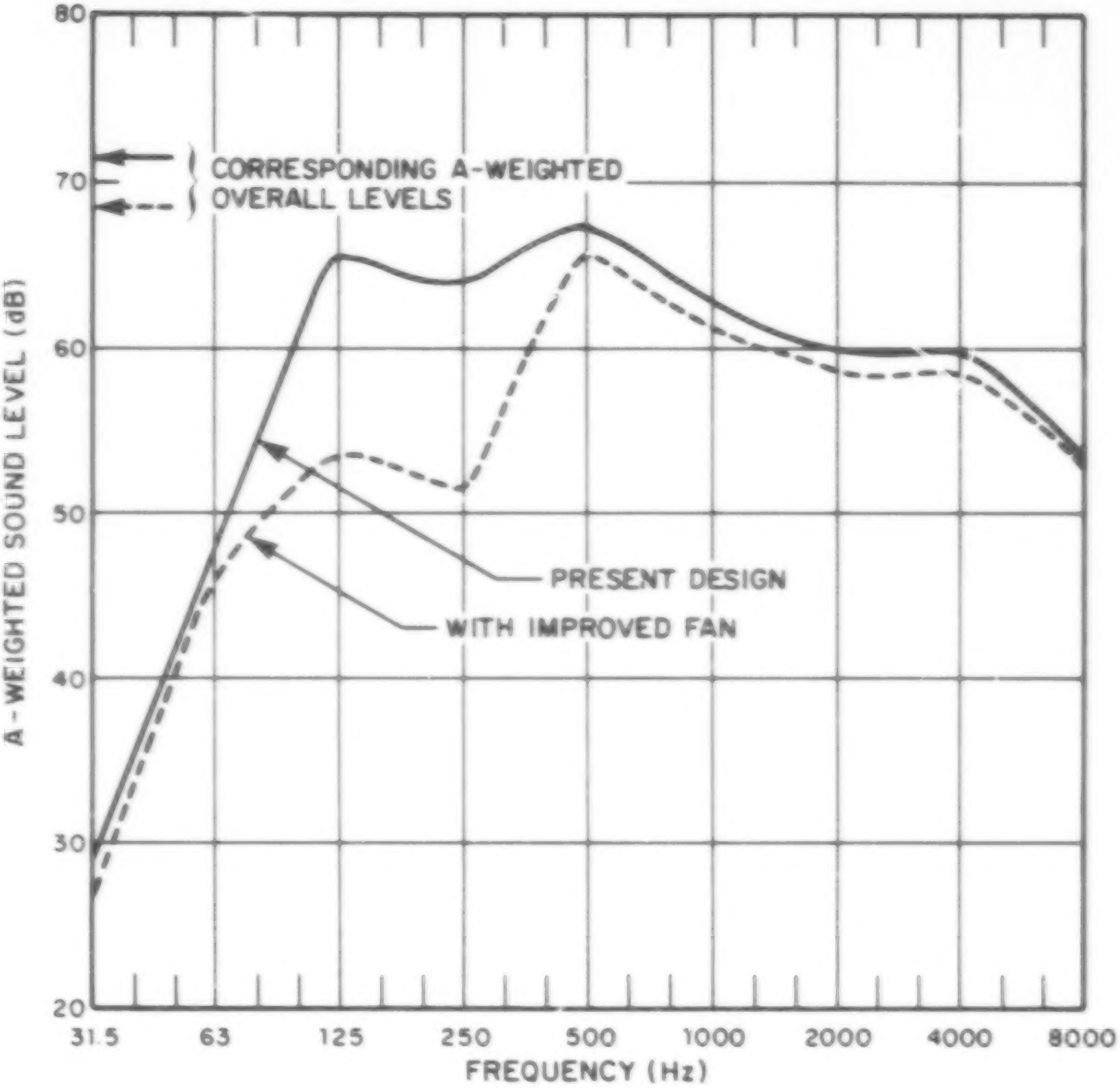


FIG. 47. A-WEIGHTED OVERALL LEVELS AND OCTAVE BAND SPECTRA SHOWING ESTIMATE OF RADIATED NOISE AT 7 m 180° FROM WORTHINGTON 750-QTEX WITH AN IMPROVED FAN DESIGN.

at the 7 m position in line-of-sight with the fan intake. To obtain the results illustrated in Fig. 47, the measured noise and measured fan contribution are used to determine the contribution from sources other than the fan. Then the original fan contribution is modified by a downward frequency shift of 57% (proportional to decrease in rpm) and a 14-dB reduction in level. This is then logarithmically combined with the nonfan contributions. The result, shown in Fig. 47, yields logarithmically an A-weighted overall level that is 3-dB down from the present level.

Improved fan design has a much smaller effect on radiation to other measurement points (e.g., off the sides), since there the fan contribution does not dominate. On the side, no more than a 1-dB reduction could be achieved and no reduction would be achieved at the front of the compressor. Nevertheless, the fan noise would have to be reduced in any systematic program of noise reduction, because, as other contributions are reduced, its contributions become more significant.

To estimate costs for the improved fan design, we need only consider the cost of introducing additional speed reduction, because the new fan itself should be no more expensive than the present one. A maximum estimate is obtained by considering that the reduction requires a reducing idler. This would require additional V-belts, a bracket to hold a bearing shaft, a set of idler pulleys with internal bearings to obtain 50% reduction, and associated assembly labor costs. We estimate this at $100 for materials and an additional 15.0 hrs of labor.

4.5.3 Exhaust-related noise reduction

Reducing engine exhaust-related noise of the present 750-QTEX unit centers primarily on the muffler shell-radiated noise. Further reduction of exhaust shell noise is possible by enclosing the muffler within an extension of the present enclosure or adding an additional sheet metal box that covers the muffler. High-temperature insensitive acoustic absorptive material such as a glass fiber must be used. Provision must also be made to supply cooling air to the muffler space, such as exhausting some of the cooling air through the space.

Quieting the exhaust-outlet noise will require technology used in other industries. Conventional means of quieting would call for a third muffler in series. A better way to quiet, especially since the major exhaust-outlet contribution is at the firing frequency and harmonics, is to use a manifold muffler to provide an expansion chamber adjacent to the engine exhaust ports (Kaye et al., 1973). This procedure will reduce the fundamental firing frequency and harmonics by about 10 dB. The manufacturer's cost for a manifold muffler is about $100 for a single manifold on a 6-cylinder in-line engine (Pax et al., 1974). For the 6V-71 engine, which has two banks of 3 cylinders, the manifold muffler would probably cost $150.

To estimate the cost of further reducing the muffler shell-radiated noise, we consider an enclosure around the present muffler. A 6-sided box of 16-gauge steel, 13.4 m^2 in total area and lined with a high temperature-resistant glass fiber, would cost approximately $80 for materials and require 10 hr of labor. Appendix B lists the individual costs to reach 65 dBA with this unit and assesses the increased consumer costs for a 65-dBA machine.

4.6 Quieting to 65 dBA

Using the noise reduction techniques discussed in Sec. 4.5, each component source level can be reduced and a 65-dBA compressor achieved. Retrofitting the present 750-QTEX design would require massive inlet and outlet cooling-air ducts, with attendant weight and size penalties. A more practical solution would be a fresh approach to the overall compressor design, using double wall enclosure construction with the cooling-air ducts built into the enclosure. Also to be considered for a cost-effective noise control design is the replacement of the present engine with an engine of lower noise emission characteristics. Table 3 lists the component source levels, for the present machine and those required for a 65-dBA machine.

TABLE 3. ADDITIONAL QUIETING OF 750-QTEX FOR 65 dBA OVERALL

Noise Source	Present Level dBA	Target Level dBA	Quieting Technique
Engine and Compressor (include air intake)	69.0	59.0	Increase noise reduction of enclosure
Engine Cooling Fan	62.5	59.5	Slower turning fan with increased blade angle
Muffler Shell	70.0	60.0	Enclose mufflers*
Exhaust Outlet	67.0	57.0	Manifold mufflers*
TOTAL	75.0†	65.0	

---

*Alternative solution is to replace the present engine with an engine having lower exhaust source levels.

†Actual measured level was 73.2 dBA, but 75 dBA is the sound level deemed characteristic by the manufacturer.

5. FOREIGN AIR COMPRESSORS

The regulation of portable air compressors in the United States will require all air compressor manufacturers to quiet at least some of their products. Air compressors presently offered as quieted units generally comply with local city codes, the New York City codes being the most stringent. Foreign noise codes are also often stringent, particularly in Europe. Consequently, European manufacturers of air compressors may be in a unique competitive position to sell quieted air compressors in the United States.

5.1 European Manufacturers

The portable air compressor market in Europe is similar to the United States market in that European domestic production competes with foreign competition, mainly American air compressor manufacturers. All of the major American air compressor manufacturers sell their products in Europe. The models they offer in Europe are similar to the ones made for the United States market.

In Britain, there are noise codes in various cities, and British manufacturers offer quieted models. CompAir, Ltd., foremost air compressor manufacturers in Britain, produce and advertise silenced versions (down to 70 dBA) of their Holman, BroomWade and Maxam compressors. Ingersoll-Rand is also building and advertising 70-dBA compressors. During compilation of BBN Report No. 2566c (the initial report on air compressors), CompAir, whose products are available in the U.S. market, was contacted through a distributor. Table 4 lists both British manufacturers and others marketing in Britain.

TABLE 4. BRITISH MAKERS OF PORTABLE AIR COMPRESSORS

1.	Atlas Copco (GB) Ltd., P.O. Box 79, Hemel Hempstead, Herts. HP2 7HA	Screw and other
2.	Bristol Pneumatic Tools Ltd., Causeway Works, Fishponds, Bristol, BS16 3JS.	Reciprocating 2-stage up to 300 cfm
3.	CompAir Construction & Mining Ltd., Holman Works, Camborne, Cornwall TR14 8DS.	Screw
	(CompAir Industrial Ltd., BroomWade Works, P.O. Box 7, High Wycombe, Bucks. HP13 5SF)	Sliding-vane rotary 2-stage
4.	Consolidated Pneumatic Tool Co. Ltd., C.P. House, 97-107 Uxbridge Road, London W5 5TP. (Chicago Pneumatic)	Sliding-vane rotary
5.	Gardner Denver (U.K.) Ltd., Brick Knoll Park, Ashley Road, St. Albans Herts. AL 1 5 UB.	Screw
6.	Hydrovane Compressor Co. Ltd., Claybrook Drive, Washford Industrial Estate, Redditch, Worcs.	Sliding-vane rotary up to 200 cfm
7.	Ingersoll-Rand Co. Ltd., Bowater House, 114, Knightsbridge, London SW1X 7LU.	Screw and other

France has a nationwide noise code based on the end use of air compressors. This information has been documented in a February 1974 Informatics, Inc. report to the EPA. In France, certification of air compressors is required. The certification is based on a 1-m sound level. We obtained the results of certification for the various manufacturers, and found that Ingersoll-Rand and Chicago Pneumatic both have certified their silenced products. A list of the domestic French manufacturers of portable air compressors is given in Table 5.

Most of the other major portable air compressor manufacturers in Europe are in the German Federal Republic, though Atlas Copco has a manufacturing plant for portable air compressors in Belgium. We contacted Dr. Rainer Prickarz of the Institute for Construction and Building in Aachen, Germany, which tests for the German Federal Republic the noise emission of portable air compressors. A report, in German, issued in September 1971 by Dr. Frenking, lists the measured sound levels for air compressors marketed in Germany; the compressors are code numbered. Discussion of the report with Dr. Prickarz indicated that the sound levels reported are the average levels measured on a 10-m radius from the geometric center of the machine. These sound levels range at full power from 66 dBA to 87 dBA.

TABLE 5. MAJOR FRENCH MANUFACTURERS OF PORTABLE AIR COMPRESSORS

1. Bernard
18 du Joliet-Curie
BP 12
69631 Venissieux

2. Paul Thomé
15-17 Chemin du Génie
69632 Venissieux

3. Ste Maco-Meudon
BP 15
69190 Genas

4. CLM Peugeot
79 Me Noel Pens
BP 35
92004 Wanterre Cede

The quietest machines in Germany are all Atlas Copco models. These models are identical to the Atlas Copco units offered in the United States. The Atlas Copco VSS-170 (Sec. 3) is listed for 67 dBA at 10 m, compared to 71 dBA when measured at a 7-m distance from the surface per ISO-2151-1972. At this time, no compressors are marketed in Germany at less than the 70 dBA per CAGI/PNEUROP or ISO-2151-1972 rating, but those in the planning stage will comply with the projected 1975 regulations. Table 6 lists the major manufacturers in Germany and Belgium.

5.2 Noise Control Technology

In Europe, the definite leader in noise control is Atlas Copco. As discussed before, measurements of portable air compressors in Europe indicate that Atlas Copco, CompAir, and Ingersoll-Rand offer units in the same "super silenced" 70-dBA category. The quieting technology used by Atlas Copco for "super silenced" compressors is documented in Sec. 3 of this report. CompAir compressors use a similar double-wall construction with

TABLE 6. BALANCE OF MAJOR EUROPEAN MANUFACTURERS

1. Atlas Copco Airpower N.V.
Boome Stenwes 957
2710 Wilrijk
Belgium

2. Bauer Tamrock-Maschinenfabrik GmbH.
8192 Gersteried 1 (066)
Postfach 370
Germany

3. Demas A.G. (DEPRAG)
Wolfgang-Rentee-Plate
41 Duisburg 1
Germany

4. Flottman-Werke GmbH.
4690 Herne
Postfach 240 and 260
Germany

5. FMA Pokorny
6000 Frankfurt (Main) 90
Postfach 900 360
Germany

6. Irme & Elze Maschinenfabrik
4970 Bad Ueynhansen
Postfach 420
Germany

cooling air ducted between the walls. The "super silenced" Atlas Copco air compressors are all reciprocating type. Discussions with Atlas Copco indicate that reciprocating air compressors are more efficient, with less heat rejection. Atlas Copco used air-cooled diesel engines with cooling fans built in, which demonstrate a much better performance than the fans we measured on domestic air compressors. CompAir and Ingersoll-Rand compressors use a sliding-vane or rotary-screw type compressor with a water-cooled Perkins or Ford diesel engine. The pusher-type fan is well shrouded. Proper air flow through either unit requires door-shut operation.

The noise control technology used in Europe is similar to that used in the United States, but a more systematic approach is applied to quieting air compressors. Noise control design is more from the frame up and uses an integrated approach rather than merely adding on quieting silencers. Foreign "super silenced" air compressors tend to have a boxy look. The outer enclosure is double walled, and serves as an air duct and silencer as well as a barrier to engine- and compressor-radiated noise. Diesel engines are used almost exclusively.

5.3 The Cost of Quieting Foreign Air Compressors

To assess the cost of quieting in foreign air compressors, we used the approach followed in the initial report on portable air compressors in the United States (BBN No. 2566c). We obtained list prices for quieted pairs of air compressors and compared the list price difference. Most interesting of these list price differences are the Atlas Copco and Ingersoll-Rand prices for their quieted and superquieted versions. Table 7 lists the price differences for these models and others. Note that the same models

TABLE 7. LIST PRICE DIFFERENCES FOR STANDARD VS QUIETED MACHINES IN EUROPE (ALL DIESEL-DRIVEN)

Manufacturer & Model	Capacity cfm	Diesel or Gasoline	List Price and Percent Increase					
			U.S.($)		France(F)		Germany(DM)	
Atlas Copco ST-35	125	Diesel	6,130		32,000			
Atlas Copco STS-35	125	Diesel	6,500	6%	35,900	12%		
Atlas Copco VSS-125	125	Diesel	6,950	13%	41,700	27%		
Atlas Copco ST-48	160	Diesel	7,190		40,100			
Atlas Copco STS-48	160	Diesel	7,500	4%	43,900	9%		
Atlas Copco VSS-170	170	Diesel	8,120	13%	50,000	25%		
Paul Thomé GD18	65	Diesel			16,020			
Paul Thomé GDI18	65	Diesel			18,325	14%		
Paul Thomé GD40	140	Diesel			32,860			
Paul Thomé GDI40	140	Diesel			39,200	19%		
Paul Thomé GD601	200	Diesel			48,120			
Paul Thomé GDI601	200	Diesel			55,030	14%		
CLM Peugeot R490	150	Diesel			35,717			
CLM Peugeot R490L	150	Diesel			41,016	15%		
CompAir CR150	150	Diesel	7,600					
CompAir CR150S	150	Diesel	8,740	15%				
CompAir CR150SS	150	Diesel	11,400	50%				
Ingersoll-Rand DR-85	85	Diesel	4,700		23,600			
Ingersoll-Rand DR-85S	85	Diesel	5,000	6%	26,900	14%		
Ingersoll-Rand DR-175	175	Diesel	7,350		44,750			
Ingersoll-Rand DR-175S	175	Diesel	7,750	5%	50,259	12%		
Ingersoll-Rand DR-2000	2000	Diesel	89,300		448,700			
Ingersoll-Rand DR-2000S	2000	Diesel	102,700	15%	541,300	20%		
Bernard	90	Diesel			21,630			
Bernard	90	Diesel			24,770	14%		
Bernard	175	Diesel			36,170			
Bernard	175	Diesel			41,590	15%		
Pokorny DV35	125	Diesel					19,580	
Pokorny SD35	125	Diesel					20,780	6%
Pokorny SSD35	125	Diesel					26,460	35%
Pokorny DV50	175	Diesel					27,020	
Pokorny SDV50	175	Diesel					29,210	8%
Pokorny SC5	175	Diesel					30,890	14%
Pokorny DW70	250	Diesel					37,810	
Pokorny SDW70	250	Diesel					40,340	6%
Pokorny SC7	250	Diesel					43,460	15%

offered in the United States and in France have a higher price difference in France. Discussion with Atlas Copco about this difference indicates that Ingersoll-Rand is the price leader in the United States. Ingersoll-Rand established price differentials that lie in the 4% to 10% range for quieted versions. Atlas Copco contends that the company sells quieted versions in the United States at a loss. Comparing the percent price increase in the United States and in France for the Atlas Copco units, we note that the price increases in Europe are almost twice as much as in the United States. This fact indicates that either Atlas Copco is making a lot more money in France or the company is discounting to market in the United States. Appendix A of this report estimates the projected customer-price increase for the VSS-170 as $1372, or $442 higher than the actual price increase calculated from Table 7. This indicates that Atlas Copco does indeed sell "super silenced" air compressors at a loss, or at least with little profit.

Comparing the two French manufacturers, Paul Thomé and CLM Peugeot, we see that they ask a 10% to 20% price increase for their silenced versions. CompAir, in marketing in the United States, listed a 15% increase for the silenced version and a 50% increase for the "super silenced" version. (These prices were estimates, since a super-silent version was in a prototype status.)

5.4 Competition by Foreign Air Compressor Manufacturers

If strict noise control becomes a reality, American manufacturers of small diesel driven compressors (less than 500 cfm) may lose the domestic market, temporarily, to their European counterparts.

American manufacturers of gasoline-driven and large diesel-driven compressors do not face the problem of foreign competition, since there is no significant production of either compressor outside the United States.

Right now, small diesel-driven compressors that meet stringent noise codes are on sale in Europe. "Super silenced" European models could easily meet an EPA-imposed noise limit, in the short run, and they could be imported more readily and cheaply than American models could be produced.

Atlas Copco and CompAir already offer 70-dBA units in the United States. Ingersoll-Rand could import quieted units from their British factory. Flottman-Werke and FMA Pokorny, both in Germany, need only a distribution network to begin exporting to the United States. (Conversations with Flottman officials indicate that a marketing arrangement is presently conducted with Worthington CEI.) Table 8 lists the addresses of the above firms, potential competitors to American manufacturers.

TABLE 8. CHIEF COMPETITORS TO AMERICAN MANUFACTURERS

Atlas Copco Airpower N.V.
Boome Stenwes 957
2710 Wilrijk
Belgium

CompAir Construction & Mining Ltd.
Holman Works
Camborne
Cornwall TR 14 8DS, England

Ingersoll-Rand Co. Ltd.
Bowater House
114, Knightsbridge
London, SWIX 7LU
England

Flottman-Werke GmbH.
4690 Herne
Postfach 240 und 260
Germany

FMA Pokorny
6000 Frankfurt (Main) 90
Postfach 900 360
Germany

Significant impact on the air compressor market by foreign models can occur only if Level 2 (70 dBA at 7 m) is adopted with less than a two- to three-year lead time for compliance. With a two-year allowance for advanced companies to quiet their air compressors and three years for small marginal companies, American manufacturers should be able to compete successfully with any European producer.

6. CONCLUSIONS

From our study of three quieted air compressors, these conclusions emerge:

- Noise control techniques vary. Only a few noise control devices or techniques are common to the three air compressors; we found that in judging the two Worthington units and one Atlas Copco unit, we were really judging the effectiveness and cost of add-on design (Worthington) vs system design (Atlas Copco).
- Regardless of design, however, all three quieted machines can be quieted even further, with little increase in fuel consumption.
- Air compressors can be quieted to EPA Levels 1 and 2 (listed in BBN Report No. 2566c) by technology already known to the air compressor industry. Quieting to Level 3 calls for noise control devices and techniques that will be new to the industry.
- Quieting to all three levels can be achieved at less cost to the consumer than was projected in the first BBN report on air compressors (see Table 9). Price increases in that report (BBN No. 2566c) were based on manufacturers' list prices.
- We do not agree with manufacturers who claim that the projection of prices for quieted air compressors (Report No. 2566c) would result in unprofitable sales.
- New and quieter designs are needed for engines used in portable air compressors, so that noise reduction goals can be achieved with a minimum of enclosing or muffling.
- Noise performance standards should be established, for both engines and muffling systems.

TABLE 9. CONSUMER COST OF QUIETING STANDARD MACHINES

Compressor Class	Based on List Price Differences (BBN Report No. 2566c) Includes 2-dBA Manufacturers Tolerance			Based on Estimated Manufacturers' Cost[†] (Appendix A and B) Includes 2-dBA Manufacturers' Tolerance		
	Gasoline	Diesel ≤500 cfm	Diesel >500 cfm	Gasoline	Diesel ≤500 cfm	Diesel >500 cfm
Level 1*	76 dBA	76 dBA	78 dBA	76 dBA	72 dBA	77 dBA
Price Increase ($/cfm)	6.11	8.40	6.22	5.98	8.08	3.69
Level 2	73 dBA	70 dBA	73 dBA	73 dBA	70 dBA	73 dBA
Price Increase ($/cfm)	12.54	21.59	9.10	8.29	10.89	5.69
Level 3	65 dBA	65 dBA	65 dBA	65 dBA	65 dBA	65 dBA
Price Increase ($/cfm)	32.10	45.19	14.18	14.14	23.06	9.69

---

*Sound level per ISO-2151-1972 at 7m on a not-to-exceed basis.

[†] Preliminary estimates: the economic impact contractor is to refine the price estimates using industry burden rates.

REFERENCES

Bender, E.K., Patterson, W.N., and Kaye, D.H. (1974). "Source Analysis and Experiments with Noise Control Treatments Applied to Freightliner Truck," BBN Report No. 2317c.

Donaldson, Inc., Acoustical Technical Bulletin, issued 15 March 1968, for Project 8918c, "Effect of Exhaust System on Engine Performance."

Fax, G.E. and Kaye, M.C. (1974). "Truck Noise III-D, The Cost of Quieting Heavy Cab-Over-Engine Diesel Tractors," BBN Report No. 2563, prepared for the Dept. of Transportation, Washington, D.C.

Freeze, T.W., Fox. H.L., and Patterson, W.N. (1974). "Portable Air Compressors: The Costs, Benefits, and Penalties of Reducing their Noise," BBN Report No. 2566c. (The current list price includes a general price increase of 5% since the data were obtained.)

Holler, H.G., Society of Automobile Engineers paper 7000535 (included in SP-359, July 1970), "The Influence of Induction and Exhaust System Design on Power Producing Characteristics of Diesel Engines."

Informatics, Inc., a draft report, issued 15 February 1974, for the Office of Noise Abatement and Control, U.S. Environmental Protection Agency, "Foreign Regulation for Construction Equipment: A Status Report."

Kaye, M.C., Patterson, W.N., and Bender, E.K. (1973). "Truck Noise III-A: Preliminary Noise Diagnosis of Freightliner Datum Truck-Tractor," prepared under contract DOT-TST-73-6.

Mason, R.L. (1974). "Progress with the Reduction of Diesel Engine Noise," *J. Sound and Vib.*

Scharton, T.D., Pinkel, B., and Wilby, J.F. (1973). "A Study of Trailing Edge Blowing as a Means of Reducing Noise Generated by the Interaction of Flow with a Surface," prepared under contract NAS1-9559 by BBN.

Thien, G.E , "Methods and Problems in Noise Reduction on High-Speed Diesel Engines," presented at SAE Meeting, Detroit, Michigan.

APPENDIX A: COST OF QUIETING

The assessment of manufacturers' costs is difficult for someone outside manufacturers' management circles. For our purposes, we will count as manufacturers' costs both the actual cost to buy an item or raw material and also the cost of floor labor to assemble this raw material into a finished product. R&D expenses are not included, since existing technology was used to quiet the compressors. Since air compressors are typically manufactured as a job-shop type of assembly rather than in a production line, we have based our raw material cost on low to medium quantity prices.

Table 10 gives a listing of the material prices we used in our assessment of manufacturers' costs. To produce a finished product at a job shop, the cost is the cost of the raw materials or OEM (Original Equipment Manufacturer) items plus the labor to fabricate and install the quieting modifications. For each of the three air compressors studied, we have assessed the cost of various quieting components. Each quieting item is listed, with the OEM or material cost and the labor required to fabricate and install.

Table 11 lists the manufacturers' cost to quiet the Worthington 160 QT Monorotor. For discussion of the noise control effectiveness of each of these items, see Sec. 2.

Table 12 lists the costs for the Atlas Copco VSS-170 unit.

Table 13 lists the costs for the Worthington 750-QTEX "Blue Brute".

TABLE 10. MATERIAL COSTS USED

Steel	
Sheet steel	$0.15 per lb
Structural steel	0.17 per lb
Perforated steel	0.30 per lb
Sound Absorptive Materials	
1-in.-thick embossed foam	0.77 per sq ft
1-in. foam w/mass barrier	1.65 per sq ft
1-in. foam	0.50 per sq ft
1-in. fiberglass	0.35 per sq ft
Vibration Damping Materials	
Spray coating	0.20 per sq ft
Gasketing	
1/2-in. × 3/8-in. foam	0.11 per linear ft
3/4-in. × 5/16-in. foam	0.16 per linear ft
Refrigerator-type door seal	0.23 per linear ft

TABLE 11. WORTHINGTON 160 QT MONOROTOR QUIETING COSTS

Item No.	Description	OEM Cost or Material Cost	Labor to Fabricate and Install (hrs)
1	Muffler	$ 61.00	0.2
2	Blowdown silencer	9.00	0.1
3	Bottom pan	11.80 steel	2.0
4	Receiver tank wrap	2.00 steel 4.00 foam	0.3
5	Cooling-air intake silencer	12.00 steel 20.00 foam	12.0
6	Enclosure over receiver tank	9.00 steel 23.00 foam	8.0
7	Cooling-air outlet duct	5.00 steel	6.0
8	Air-intake muffler	25.00	0.1
9	Door seals	4.50	0.2
10	Larger tires	40.00	0.2
	Gross Cost	$226.30	29.1
	Less Standard Items		
	Standard muffler	-25.00	0.1
	Standard tires	-30.00	0.2
	Net Cost	$171.30	28.8

TABLE 12. ATLAS COPCO VSS-170 SUPER SILENCED MODEL QUIETING COSTS

Item No.	Description	OEM Cost of Material Cost	Labor to Fabricate and Install (hrs)
1	Outer enclosure and duct work	$ 96.00 steel 58.00 foam 15.00 fiberglass 30.00 damping 24.00 perf. steel 10.00 gasketing	50.0
2	Under pan	17.50 steel	10.0
3	Doors	4.50 steel 3.00 foam 6.00 perf. steel 7.00 gasketing	10.0
4	Muffler wrapping	7.00 fiberglass 3.00 steel	1.0
5	Larger receiver tank	100.00	1.0
	Gross Cost	$376.00	73.0
	Less Standard Items		
	Standard receiver tank	70.00	1.0
	Regular enclosure	57.00 steel	10.0
	Net Cost	$249.00	62.0

TABLE 13. WORTHINGTON 750-QTEX "BLUE BRUTE" QUIETING COST

Item No.	Description	OEM Cost of Material Cost	Labor to Fabricate and Install (hrs)
1	2 large mufflers	$325.00	3.0
2	Jacketing of one muffler	14.00 steel 12.00 fiberglass	2.0
3	Blowdown muffler	13.00	0.1
4	Quiet sphere flex. connector	103.00	0.2
5	1¼-in. foam w/inter-spaced mass wall 8 doors plenum others	 60.00 49.00 13.00	 1.0 0.7 0.3
6	Gaskets 1/2 in. × 3/8 in. 3/4 in. × 5/16 in.	 7.15 7.68	 0.5 0.3
7	Perforated door covers	29.00	2.0
8	Floor pan	45.00	20.0
9	Cooling-air intake silencer	35.00 steel 30.00 foam 40.00 perf. metal	30.0
10	Side cooling-air exhaust silencer (2)	9.00 steel 12.00 foam 28.00 perf. metal	26.0
11	Top cooling-air exhaust system	5.00 steel 7.00 foam 17.00 perf. metal 10.00 air cyl.	20.0
12	Cooling fan	100.00	0.5
	Gross Cost	$880.83	106.6
	Less Standard Items		
	Standard muffler	100.00	1.0
	Blowdown muffler	13.00	0.1
	Quiet sphere flex. connector	103.00	0.2
	Net Cost	$654.83	105.3

To compare these manufacturing costs to our earlier report (BBN 2566c), which used customer list prices, we have marked these costs up by certain multipliers for OEM cost, shop labor, and distributor markup. These multipliers are based on markups observed in other industries. Table 14 gives the multipliers that we used for list price comparison, i.e., a 1.5 multiplier for all OEM items or any material bought to quiet compressors; $12 per hour for shop labor; and a 23% margin as a distributor's markup. Note that these costs do not include any R&D expenses beyond normal engineering overhead on shop labor; this is merely the cost to fabricate and sell the item. These calculations to derive customer price increases based on manufacturers' costs are, at best, first cuts for comparison only.

TABLE 14. MULTIPLIERS FOR LIST PRICE COMPARISON

Description	Multiplier
OEM Cost Markup	1.5
Shop Labor	$12.00/hr
Distributors' Markup	1.23 (23%)

No provision is made for R&D costs other than the overhead on shop labor. These multipliers are based on our experience in working with other industries and may not be characteristic of the compressor industry.

Table 15 gives the calculations of the cost of quieting for the Worthington 160 QT Monorotor, a gasoline-driven air compressor. The net customer price increase should be $742 for 14 dBA of noise reduction. This gives a marginal cost of $.33 per cfm per dBA of noise reduction, or a net price difference of $4.64 per cfm. Comparison is also given for the BBN Report No. 2566c for gasoline-driven compressors, which says that, based on list price differences, the average marginal cost charged by air compressor manufacturers is $.61 per cfm per dBA and that the net price difference to get to Level 1 or 76 dBA for this product – with no tolerance – is $4.15 per cfm, or – with tolerance – $6.11 per cfm. The calculated price increase required to quiet this unit, $4.64 per cfm, brackets the last two values.

Table 16 gives a calculation of the cost of quieting for the Atlas Copco VSS-170 diesel-driven unit less than 501 cfm. The noise reduction for this unit was 12 dBA; the projected customer price increase, $1,372. This gives a marginal cost of $.67 per cfm per dBA or a net price difference of $8.08 per cfm. This unit falls within the Level 1 unit with tolerance and could be sold as a Level 2 unit. According to BBN Report No. 2566c for diesel-driven compressors less than 501 cfm, the average marginal cost of quieting in this range would be $.61 per cfm per dBA for quieting to 76 dBA. When quieting in the range of 76 to 70 dBA, the marginal cost is $1.60 per cfm per dBA. The net price difference for Level 1 with no tolerance is $6.23 per cfm. With the 2-dBA manufacturing tolerance, it is $8.44 per cfm. The net price difference calculated for this model brackets the two numbers. However, this is actually a Level 2 product, which with no tolerance should have a net price difference of $15.79 per cfm.

TABLE 15. CALCULATION OF MARGINAL COST OF QUIETING (CONSUMER) FOR WORTHINGTON 160 MONOROTOR (GASOLINE-DRIVEN)

Level 1 Product		
Noise Reduction (ISO-2151-1972) 14 dBA, i.e., 89 dBA for Standard Unit, 75 dBA for Quieted Version		
Net Cost	$171.30 material	28.8 hrs
Dealer Price	(1.5)($171.30) = $257.50	($12/hr)(28.8 hrs) = $346.00
Σ	$257.50 + $346.00 = $603.50	
Customer Price Increase (no R&D expenses)	(1.23)($603.50) = $742.00	
Marginal Cost	$\frac{(\$742.00)}{(160\text{ cfm})(14\text{ dBA})}$ = $0.33/cfm/dBA	
Net Price Difference	$\frac{\$742.00}{160\text{ cfm}}$ = $4.64/cfm	
From BBN Report No. 2566c — Gasoline-Driven Compressors		
Marginal Cost		$0.61/cfm/dBA
Net Price Difference	No tolerance w/2-dBA tolerance	$4.15/cfm $6.11/cfm

TABLE 16. CALCULATION OF MARGINAL COST OF QUIETING (CONSUMER) FOR ATLAS COPCO VSS-170 Dd (DIESEL-DRIVEN ≤ 500 cfm)

Level 1 or 2 Product		
Noise Reduction (ISO-2151-1972) 12 dBA, i.e., 82 dBA for Standard Unit, 70 dBA for Quieted Version		
Net Cost	\$249.00 material	62.0 hrs
Dealer Price	(1.5)(\$249.00) = \$374.00	(\$12/hr)(62.0 hrs) = \$744.00
Σ	\$374.00 + \$744.00 = \$1,118.00	
Customer Price Increase (no R&D expenses)	(1.23)(\$1,118.00) = \$1,372.00	
Marginal Cost	$\frac{(\$1,372.00)}{(170\ \text{cfm})(12\ \text{dBA})} = \$0.67/\text{cfm}/\text{dBA}$	
Net Price Difference	$\frac{\$1,372.00}{170\ \text{cfm}} = \$8.08/\text{cfm}$	
From BBN Report No. 2566c — Diesel-Driven Compressors ≤ 500 cfm		
Marginal Cost of Quieting	Level 1 (to 76 dBA) \$0.61/cfm/dBA Level 2 (76 dBA to 70 dBA) \$1.60/cfm/dBA	
Net Price Difference	Level 1 no tolerance w/2-dBA tolerance Level 2 no tolerance	\$ 6.23/cfm 8.44/cfm \$15.79/cfm

TABLE 17. CALCULATION OF MARGINAL COST OF QUIETING (CONSUMER) FOR WORTHINGTON 750-QTEX "BLUE BRUTE" (DIESEL-DRIVEN > 500 cfm)

Level 1 Product		
Noise Reduction (ISO-2151-1972) 13 dBA, i.e., 89 dBA for Standard Unit, 75 dBA for Quieted Version		
Net Cost	$654.83 material	105.3 hrs
Dealer Price	(1.5)($654.83) = $980.00	($12/hr)(105.3 hrs) = $1,264.00
Σ	$980.00 + $1,264.00 = $2,244.00	
Customer Price Increase (no R&D expenses)	(1.23)($2,244.00) = $2,765.00	
Marginal Cost	$\frac{(\$2,765.00)}{(750\ \text{cfm})(13\ \text{dBA})} = \$0.28/\text{cfm/dBA}$	
Net Price Difference	$\frac{(\$2,765.00)}{(750\ \text{cfm})} = \$3.69/\text{cfm}$	
From BBN Report No. 2566c — Diesel-Driven > 500 cfm		
Marginal Cost		$0.36/cfm/dBA
Net Price Difference	no tolerance w/2-dBA tolerance	$5.38/cfm $6.22/cfm

APPENDIX B: PROJECTED COSTS TO REDUCE COMPRESSOR NOISE TO 65 dBA

In this appendix, we assess the manufacturers' cost of quieting to 65 dBA the three compressors we studied. As in Appendix A, we estimate both material and additional shop labor required to quiet each component and produce the total material and hour cost to the manufacturer.

We also add research and development costs that we believe are necessary. The component quieting is referenced back to each respective section. Summaries of manufacturers' costs and customer list price increases are given in Tables 18 through 20.

The Worthington 160 QT machine, a gasoline-driven compressor, requires extensive redesign to reach the 65-dBA level (see Sec. 2 for redesign information). Table 18 lists each component redesign for the 160 QT by item number, with material costs and additional shop labor needed to assemble or fabricate these items. Research and development costs are estimated at $10,000 for this unit, representing a complete source identification and ranking program, including the development of noise control methods to quiet the unit.

To calculate customer list price, we amortize the R&D costs, assuming a production of 100 units per year over three years. The net cost per unit is $34. Using the cost multipliers from Appendix A, and tacking a 1.5 multiplier to the R&D cost, we arrive at a dealer price of $1047. With a 23% dealer price markup, the total customer list price increase is $1288. The marginal cost for 11-dBA reduction is $.73 per cfm per dBA, or a net price difference of $8.04 per cfm to quiet the Worthington 160 QT unit to 65 dBA.

TABLE 18. QUIETING THE WORTHINGTON 160 QT UNIT TO 65 dBA

Item	Description	Material	Additional Shop Labor
1	Vibration isolation	$ 50.00	12.0 hrs
2	Additional transmission loss on doors	55.00	10.0 hrs
3	New cooling fan shroud	100.00	10.0 hrs
4	New fan blade	25.00	
5	Change fan pulley	10.00	0.5 hrs
6	Lag muffler shell	25.00	2.0 hrs
7	New muffler	50.00	0.5 hrs
8	Redesigned cooling-air silencer	65.00	0.5 hrs
	Total	$380.00	35.5 hrs

Research and Development Costs	$10,000.00
Assumed Production Per Year	100 units
Amortize R&D Costs Over 3 Years, Cost Per Unit	34.00
Use Multipliers From Appendix A, (1.5)(380) =	570.00
($12/hr)(35.5 hrs) =	426.00
Σ	996.00
R&D Costs (1.5)	51.00
Dealer Price	$ 1,047.00
Customer List Price Increase	1,288.00
Marginal Cost (10-dBA reduction)	0.80/cfm/dBA
Net Price Difference	8.04/cfm

TABLE 19. QUIETING THE ATLAS COPCO VSS-170 UNIT TO 65 dBA

Item	Description	Material	Additional Shop Labor
1	Laminated exterior panels (substituted for existing panels)	$400.00	10 hrs
2	Acoustic absorptive treatment	120.00	15 hrs
3	Mount isolators	100.00	2 hrs
4	Additional muffler	100.00	2 hrs
	Total	$720.00	29 hrs

Research and Development Costs	$10,000.00
Assumed Production Per Year	100 units
Amortize R&D Costs Over 3 Years, Cost Per Unit	34.00
Use Multipliers From Appendix A, (1.5)(720) =	1,080.00
($12/hr)(29 hrs) =	348.00
Σ	1,428.00
R&D Costs (1.5)	51.00
Dealer Price	$ 1,479.00
Customer List Price Increase	1,820.00
Marginal Cost (5-dBA reduction)	2.14/cfm/dBA
Net Price Difference	10.70/cfm

Less extensive treatment is required to quiet the Atlas Copco unit. Exterior panels must be laminated for damping, however, and laminating is an expensive process. We estimate that the net material cost would be $400 more than present sheet metal panels. Research and development costs are estimated to be $10,000. Production per year of 100 units over a three-year period gives a net R&D cost of $34. Applying the previous cost multipliers, the dealer price is $1479 and the customer list price increase is $1820. The marginal cost of quieting from 70 to 65 dBA is $2.14 per cfm per dBA, or a net price difference of $10.70 per cfm for the Atlas Copco unit (see Table 19).

Table 20 lists the costs of quieting the Worthington 750-QTEX, another unit in which quieting, though not extensive, did require major material changes in the enclosure and, as a result, additional labor. Research and development costs are estimated at $15,000 for this unit because the scale of both unit and engine is larger than that of the other two units. R&D cost of 20 units per year over a three-year period is $250 per unit. Again, using the multipliers from Appendix A, we estimate a dealer price increase of $3347 and a total customer list price increase of $4120. For a 10-dBA noise reduction, this is a marginal cost of $.55 per cfm per dBA or a net price difference of $5.50 per cfm.

In Table 21, we compare quieting costs of all three machines, with the average values extrapolated from BBN Report No. 2566c, the first report on the cost of quieting air compressors. Appendix A lists $4.64 per cfm as the net price difference for quieting the Worthington 160 QT machine to 76 dBA. Reducing the sound level to 65 dBA at 7 m would require an additional $8.04 per cfm. The total price to reduce a standard machine to 65 dBA would be $12.68 per cfm. BBN Report No. 2566c lists the projected price as $32.12 per cfm. The difference is $19.44 per cfm.

TABLE 20. QUIETING THE WORTHINGTON 750-QTEX UNIT TO 65 dBA

Item	Description	Material	Additional Shop Labor
1	Improve enclosure	$565.00	100.8 hrs
2	Lower fan speed	100.00	15.0 hrs
3	Enclose mufflers	80.00	10.0 hrs
4	Manifold mufflers	150.00	10.0 hrs
	Total	$895.00	135.8 hrs

Research and Development Costs	$15,000.00
Assumed Production Per Year	20 units
Amortize R&D Costs Over 3 Years, Cost Per Unit	250.00
Use Multipliers From Appendix A,	
(1.5)(895) =	1,342.00
($12/hr)(135.8 hrs) =	1,630.00
Σ	2,972.00
R&D Costs (1.5)	375.00
Dealer Price	$ 3,347.00
Customer List Price Increase	4,120.00
Marginal Cost (10-dBA reduction)	0.55/cfm/dBA
Net Price Difference	5.50/cfm

TABLE 21. COMPARISON OF QUIETING COSTS FOR 65-dBA MACHINES

	Worthington 160 QT	Atlas Copco VSS-170	Worthington 750-QTEX
Net Price Difference	for 75 dBA $ 4.64/cfm to 65 dBA $ 8.04/cfm	for 70 dBA $ 8.08/cfm to 65 dBA $10.70/cfm	for 75 dBA $ 3.69/cfm to 65 dBA $ 5.50/cfm
(A) Total price increase for 65-dBA machine	$12.68/cfm	$18.78/cfm	$ 9.19/cfm
(B) From BBN Report No. 2566c (no tolerance)	$32.12/cfm	$36.79/cfm	$12.90/cfm
Difference A–B	-$19.44/cfm	-$18.01/cfm	-$ 3.71/cfm

This means that the estimated costs, using the Worthington 160 QT machine as a base, are less than half the projected costs, using the average values of price increases charged by manufacturers for comparable machines.

Similar analyses are shown in Table 21 for the Atlas Copco unit and the Worthington 750-QTEX unit. Again, the estimated costs of achieving a 65-dBA level are consistently less than the costs projected in BBN Report No. 2566c.

Using the Worthington 160 QT, the Atlas Copco VSS-170 Dd, and the Worthington 750-QTEX as data bases, and working with average price differentials, we found that the estimated cost of quieting the average air compressor is considerably below the cost originally projected.

We conclude, then, that we cannot agree with manufacturers who claim that the price increase given in BBN Report No. 2566c for quieted air compressors results in an unprofitable sale.

APPENDIX C: COMPARISON OF OPERATIONAL PERFORMANCE

For additional insight on the influence of noise control on fuel consumption, cooling performance, and pumping capacity, we compare the engine horsepower (which gives an estimate of the power input), the compressor type, and the pumping capacity of the three units tested. The comparison gives a gross indication of the relative efficiency of the three compressor types. Table 22 gives a listing of the performance parameters.

Of particular interest is the last item: fuel efficiency. The Worthington 160 QT machine, according to the manufacturer, consumes 4.57 gal/hr. The manufacturer says this fuel consumption is considered typical of all three Worthington 160 cfm units (gasoline models). The Worthington 160 QT, then, has a fuel efficiency of 2,100 cu ft of free air per gallon of fuel. The Atlas Copco unit delivers 5,100 cu ft of free air per gallon of fuel and the larger Worthington diesel-driven unit gives 4,500 cu ft of air per gallon of fuel. Neither manufacturers nor we can measure directly the difference in fuel consumption between so-called standard and quieted (Level 1) compressors. The variance between units was greater than the variance for additional quieting.

The comparative inefficiency of the small Worthington unit can be partly attributed to the inefficiency of the sliding-vane type compressor. We do not know the relative degrees of efficiency between sliding-vane, reciprocating, and rotary-screw compressors, though industry observations are that the reciprocating type is the most efficient compressor for input horsepower, followed by rotary-screw and then by sliding-vane.

Preceding page blank

TABLE 22. LISTING OF PERFORMANCE PARAMETERS.

Class (Per BBN Report No. 2566c)	Worthington 160 QT Quieted Gasoline	Atlas Copco VSS-170 Quieted Diesel ≤500 cfm	Worthington 750-QTEX Quieted Diesel >500 cfm
Engine Horsepower	65 hp	47 hp	210 hp
Compressor Type	Sliding Vane	Reciprocating	Rotary-Screw
Pumping Capacity	160 cfm @ 100 psig	170 cfm @ 100 psig	750 cfm @ 100 psig
Cooling Media	Water	Air	Water
Fan Type	Shrouded Axial	Ducted Axial and Centrifugal	Shrouded Axial
Cooling Ability (Max. Ambient Temperature) for continuous operation	110°F	105°F	122°F
Fuel Consumption	4.57 gal/hr	2.0 gal/hr	10.0 gal/hr
Fuel Efficiency	2,100 cu ft/gal @ 100 psi	5,100 cu ft/gal @ 100 psi	4,500 cu ft/gal @ 100 psi

One primary power requirement is for the cooling fan. The Worthington unit, which has the fan operating in a stalled mode, requires 8 to 9 hp of the 65-hp engine just to drive the fan. We cannot measure the Atlas Copco unit, but the fan assemblies seem to be quite efficient in pumping air, in that they use a ducted axial fan and a centrifugal fan on a flywheel. The larger Worthington unit uses a very large slow-turning fan; it is more efficient than the one in use on the Worthington 160 QT unit because it requires only 11 hp for operation. Quieting the Worthington 160 QT unit further with improved shrouding design and a larger, slower fan should reduce the horsepower required to drive the fan; fuel consumption may decrease or, conversely, fuel efficiency may increase. We believe that many of the present compressors may be operating with extremely inefficient fans, such as we found on the Worthington unit. Our research shows that if manufacturers increase pumping efficiency to reduce noise radiated by these fans, they will quite possibly cut fuel consumption as well.

GLOSSARY OF TECHNICAL TERMS

A-Weighted Sound Level

The A-weighting network of a sound level meter is a standardized frequency-dependent filter which discriminates against frequencies below 500 cycles per second and above 10,000 cycles per second. The A-weighted sound level is the single number that one would read from a sound level meter with the A-weighting network. The unit of measure is the decibel, and, to note the A-weighting, it is abbreviated dBA. Tables for converting octave band and one-third octave-band levels to an equivalent A-weighted level are given in the General Radio *Handbook of Noise Measurements*.

Acoustic Absorption

The change of sound energy into some other form, usually heat, in passing through a medium or on striking a surface.

Acoustic Power

See sound power, sound power level.

Addition, Logarithmic vs Arithmetic

(Also see subtraction.) When two sound pressure levels or sound power levels are present in the same space or at the same location, the procedure for finding the composite sound pressure or sound power level involves logarithmic addition. In this case, it is the sound *pressures* or sound *powers* that add arithmetically while their *levels* add logarithmically. To distinguish the two operations, the term "add" is used when values are to be added arithmetically and the term "combine" is used when values are to be added logarithmically. The process of logarithmic addition (combining decibel quantities) has been reduced to the following steps.

1. Determine the difference (in dB) between the two levels to be combined.

2. For this difference find the appropriate correction value (in dB) in the table below.

3. *Add* (arithmetically) this correction value to the higher of the two levels being combined. The resulting sum is the value of the combined levels.

Preceding page blank

If Difference Between Levels to be Combined Is	Then Add to the Higher Level the Following Number of dB
0 – 1 dB	3 dB
2 – 3 dB	2 dB
4 – 9 dB	1 dB
more than 9 dB	0 dB

When more than two levels are to be combined, combine the two highest levels by the procedure above. Then combine the next highest level with the level of the combination of the two highest values. Continue in this manner until all levels have been combined or all remaining uncombined levels differ by more than 9 dB from the combination of all higher levels.

Air-to-Boil (ATB)

An engineering term used to indicate the ambient air temperature where continuous operation of the compressor would result in a bulk water temperature in the radiator top tank of 212°F.

Amplitude

The maximum value of the displacement of a wave or other periodic phenomenon from a reference position.

Analyzer

A combination of a filter system and a system for indicating the relative energy that is passed through the filter system. The filter is usually adjustable so that the signal applied to the filter can be measured in terms of the relative energy passed through the filter as a function of the adjustment of the filter response-vs-frequency characteristic. This measurement is usually interpreted as giving the distribution of energy of the applied signal as a function of frequency.

Anechoic Room (Free-Field Room)

A room whose boundaries absorb effectively all the sound incident thereon, thereby affording essentially free-field conditions.

Bandwidth

See frequency band.

CFM

Cubic feet per minute.

Damping

The dissipation of energy with time or distance.

Decibel (dB)

A dimensionless unit of measure of the ratio of two quantities, each of which is equal to or proportional to power. If P_1 and P_2 are 10 times the logarithm tc the base 10 of the ratio of two such quantities, then P_1/P_2 has the dimensions of decibels.

Diffuse Sound Field

A sound field in which the time average of the mean-square sound pressure is everywhere the same and the flow of energy in all directions is equally probable.

Direct Sound Field

A sound field in which energy is flowing outward from the source. The sound field very close to a source, even in a reverberant room, is a direct field. Sound fields outdoors are direct fields at all distances from the source and are referred to as "free sound fields" or "free fields."

Directivity

The directivity of a source describes the extent to which the source radiates more sound in some directions than others.

Effective Sound Pressure (Root-Mean-Square Sound Pressure)

At a point, it is the root-mean-square value of the instantaneous sound pressures, over a time interval at the point under consideration. In the case of periodic sound pressures, the interval must be an integral number of periods or an interval long compared to a period. In the case of nonperiodic sound pressures, the interval should be long enough to make the value obtained essentially independent of small changes in the length of the interval. Note: the term "effective sound pressure" is frequently shortened to "sound pressure."

Far Field and Near Field (Acoustic)

The acoustic fields are described in terms of wave propagation. The near field is close to the source; how far it extends depends on the wavelength of the radiated sound. Normally, the acoustic near field extends to a distance of about one quarter of a wavelength. Sound pressure fluctuations within this distance correspond to the hydrodynamic response of the fluid to the motion of the adjacent surface. In the far field, the sound pressure fluctuations are caused by the propagation of sound waves away from the source. All measurements of sound power and source acoustic characteristics must be made in the acoustic far field.

Far Field and Near Field (Geometric)

Both terms describe the radiation field of a typical noise source. The *near field* (Fresnel zone) is in the vicinity of the source; how far it extends depends on the frequency, on a characteristic source dimension, and on the phases of the radiating parts of the surface. In the *far field* (Fraunhofer zone), the sound pressure level decreases 6 dB for each doubling of distance measured from the acoustic center.

Flanking Path

A path by which sound is transmitted through structural elements common to the source and receiving room, or through gaps around the perimeter of the panel, or through cracks and openings in the panel.

Free Sound Field (Free Field)

A field in a homogeneous, isotropic medium that is free from boundaries.

Frequency

The time rate of repetition of a periodic phenomenon. The frequency is the reciprocal of the period of a pure tone. Frequency is given in Hertz (Hz), equivalent to cycles per second (cps).

Frequency Band

An interval of the frequency spectrum defined between upper and lower "cut-off" frequencies. The band may be described in terms of these two frequencies, by the width of the band, and by the geometric mean frequency of the upper and lower cut-off frequencies, e.g., "an octave band centered at 500 Hz."

Gaussian Distribution (Normal Distribution)

A particular amplitude distribution of fundamental importance in the theory of probability. Its histogram is the familiar "bell-shaped curve." It describes many natural phenomena, and most stationary acoustic noise that is not periodic has an essentially Gaussian distribution.

Hard Surface

A surface which is a poor absorber of sound.

Harmonic

(1) An integral multiple of a given frequency; a sinusoidal component of a periodic wave. (2) A signal having a frequency which is a harmonic (sense 1) of the fundamental frequency.

Hearing Threshold Level (of an Ear)

Amount in decibels by which the threshold of audibility for that ear exceeds a standard audiometric threshold.

Histogram

Graph of an amplitude density distribution.

Impact

A single collision of one mass in motion with a second mass which may be either in motion or at rest.

Impedance

The mechanical impedance is the complex ratio of the force acting on a specified area of an acoustic medium or mechanical device to the resulting linear velocity through or of that area, respectively. The unit is N-sec/m in the mks system or lb-sec/ft in the English system.

Isolation

Isolation is a reduction in the capacity of a system to respond to an excitation, attained by the use of a resilient support.

Level

In acoustics, the level of a quantity is the logarithm of the ratio of that quantity to a reference quantity of the same kind. The base of the logarithm, the reference quantity, and the *kind* of level must be specified. Examples are sound power levels, sound pressure levels, and acceleration levels. Levels are always expressed in decibels (dB).

Masking

1. The *process* by which the threshold of audibility for one sound is raised by the presence of another (masking) sound.
2. The *amount* by which the threshold of audibility of a sound is raised by the presence of another (masking) sound. The unit customarily used is the decibel.

Microphone

An electroacoustic transducer that responds to sound waves and delivers essentially equivalent electric waves.

Near Field

See far field and near field.

Noise

Any undesired sound or any erratic, intermittent, or statistically random oscillation.

Noise Level

1. The level of noise, the type of which must be indicated by further modifier or context.

Note: The physical quantity measured (e.g., voltage), the reference quantity, the instrument used, and the bandwidth or other weighting characteristic must be indicated.

2. For airborne sound, unless specified to the contrary, noise level is the weighted sound pressure level called sound level; the weighting must be indicated.

Noise Reduction

The difference in decibels between the sound pressure level on either side of a boundary construction is defined as the noise reduction (NR). Unlike transmission loss, noise reduction is dependent not only on the intervening structure but also the source and receiver room acoustic properties. Noise reduction can also refer to the number of dB by which sound pressure levels are reduced in a single space or at a fixed location due to introduction of noise control treatments.

Octave

The frequency interval in Hz or cycles per second between two sounds having a frequency ratio of two.

Octave Band

A frequency range bounded by upper and lower frequency limits f_u and f_l, where $f_u = 2f_l$. Octave bands are usually specified by their geometric mean frequency called the band center frequencies. The standard octave bands covering the audible range are designated by the following center frequencies: 31.5, 63, 125, 250, 500, 1000, 2000, 4000, 8000, and 16,000 cycles per second (Hz). The corresponding upper and lower frequencies are 22/45, 45/89, 89/177, 177/354, 354/708, 708/1416, 1416/2832, 2832/5664, 5664/11,328, 11,328/22,656.

OEM

Original equipment manufacturer.

Oscillation

The variation, usually with time, of the magnitude of a quantity with respect to a specified reference when the magnitude is alternately greater and smaller than the reference.

P

Pascal, N/m^2 pressure.

Pitch

That attribute of auditory sensation in terms of which sounds may be ordered on a scale extending from low to high. Pitch depends primarily upon the frequency of the sound stimulus, but it also depends upon the sound pressure and wave form of the stimulus. Note: The pitch of a sound may be described by the frequency or frequency level of that simple tone, having a specfied sound pressure level, which is judged by listeners to produce the same pitch.

Power Level

In decibels, it is 10 times the logarithm to the base 10 of the ratio of a given power to a reference power. The reference power must be indicated.

Pure Tone

Sound wave whose sound pressure at a point varies sinusoidally with time and is characterized by its singleness of pitch. The time interval for one complete cycle of the sinusoidal pressure variation is known as the period.

PWL

(See Sound Power Level.)

Radiation Efficiency

The radiation efficiency of a vibrating surface is proportional to the acoustic power radiated per unit surface area per unit of mean-square velocity of vibration averaged over the radiating surface. It is the measure of the efficiency with which a given surface converts vibratory energy to acoustic energy.

Random Noise

An oscillation whose instantaneous magnitude is not specified for any given instant of time. The instantaneous magnitudes of a random noise are specified only by probability distribution functions giving the fraction of the total time that the magnitude, or some sequence of magnitudes, lies within a specified range. Note: A random noise whose instantaneous magnitudes occur according to Gaussian distribution is called Gaussian random noise.

Rate of Decay

The time rate at which the sound pressure level (or other stated characteristic) decreases at a given point and at a given time. A commonly used unit is the decibel per second.

Resonance

(1) The phenomenon of amplification of a free wave or oscillation of a system by a forced wave or oscillation of exactly equal period. The forced wave may arise from an impressed force upon the system or from a boundary condition. The growth of the resonant amplitude is characteristically linear in time. (2) Of a system in forced oscillation, the condition which exists when any change, however small, in the frequency of excitation causes a decrease in the response of the system.

Reverberation

1. The persistence of sound in an enclosed space, as a result of multiple reflections after the sound source has stopped.

2. The sound that persists in an enclosed space, as a result of repeated reflection or scattering, after the source of sound has stopped.

Note: The repeated reflections of residual sound in an enclosure can alternatively be described in terms of the transient behavior of the modes of vibration of the medium bounded by the enclosure.

Reverberant Field

That part of the radiated sound field where the waves reflected from the boundaries of the enclosure are superimposed upon the incident field. The reverberant field may be called a diffuse field if a great many reflected wavetrains cross from all possible directions and the sound energy density is very nearly uniform throughout the field.

Reverberation Room

A room in which all boundaries are hard and the sound field is reverberant over nearly the entire room volume with the exception of a small region around the source.

Room Constant

A value equal to the product of the average absorption coefficient of the room and the total internal area of the room divided by the quantity one minus the average absorption coefficient.

Soft Surface

A surface which is a good absorber of sound.

Sound

A disturbance that propagates through an elastic material (air, water) at a speed characteristic of that medium. The physical quantities of most interest are the sound pressure and sound power.

Sound Energy

The energy which sound waves contribute to a particular medium.

Sound Field

A region containing sound waves. Sound fields are typically described as reverberant, direct, diffuse, etc.

Sound Intensity

The average rate of sound energy transmitted in a specified direction through a unit area normal to this direction at the point considered (watts/m^2).

Sound Power

The sound power of a source is the total sound energy radiated by the source per unit of time.

Sound Power Level

PWL, in dB, is 10 times the logarithm to the base 10 of the ratio of a given sound power to a reference sound power. (Standard reference is 10^{-12} watts.)

$$\mathrm{PWL} = 10 \log\left(\frac{W \text{ (watts)}}{10^{-12} \text{ watts}}\right) \text{ dB re } 10^{-12} \text{ watts} .$$

Sound Pressure

The total instantaneous pressure at a point in the presence of a sound wave minus the static pressure at that point.

Sound Pressure Level

SPL, in dB, is 20 times the logarithm to the base 10 of the ratio of the pressure of this sound to a reference pressure. (Standard reference is 20 $\mu N/m^2$ for airborne sound. Different reference pressures are sometimes used for underwater sound.)

$$\text{SPL} = 20 \log\left(\frac{p}{0.0002\ \mu\text{bar}}\right) \text{dB re } 20\ \mu\text{P} .$$

Spectrum

(1) The spectrum of a function of time is a description of its resolution into components, each of different frequency and (usually) different amplitude and phase. (2) Also used to signify a continuous range of components, usually wide in extent, within which waves have some specified common characteristic: e.g., "audio-frequency spectrum."

Stationary (Statistical)

Term that describes a noise whose spectrum and amplitude distribution do not change with time.

Subtraction, Logarithmic

The procedure for logarithmic subtraction is more complicated and less precise than the procedure for addition, since it is not known beforehand whether the contribution being determined is larger or smaller than the combination of all the other contributions. Consider the composite level, L_{comb} (in dB), to be the combination of two levels L_1 and L_2 (both in dB). If L_{comb} and L_1 are known, then L_2 can be found by the following method.

Subtracting Decibels

If L_{comb} Minus L_1 is	Then L_2 is Less Than L_{comb} by
0 dB	more than 10 dB
1 dB	5 – 10 dB
2 dB	4 – 5 dB
3 dB	3 – 5 dB
4 dB	2 – 3 dB
5 dB	1 – 2 dB
6 – 10 dB	1 dB
more than 10 dB	0 dB

Tone

(1) A sound wave capable of exciting an auditory sensation having pitch.

(2) A sound sensation having a pitch.

Transmission Coefficient

The sound transmission coefficient of a partition is the ratio of transmitted sound intensity to the incident sound intensity.

Transmission Loss

The sound transmission loss (TL) of a partition, in dB, is the difference between incident sound intensity level and transmitted sound intensity level. It can be expressed as 10 times the logarithm to the base 10 of the reciprocal of the transmission coefficient, τ.

$$TL = 10 \log \frac{1}{\tau} = -10 \log (\tau) \text{ dB} .$$

Vibration

Motion due to a continuous change in the magnitude of a given force which reverses its direction with time.

Vibration Isolator

A resilient support that tends to isolate a system from steady-state excitation.

Vibration Level

Vibration may be described by many parameters, including displacement, velocity,or acceleration, each in a particular direction. Of these, acceleration is the easiest to measure; thus, it is most commonly used.

Wave

A disturbance which is propagated in a medium in such a manner that at any point in the medium the quantity serving as a measure of the disturbance, such as pressure or particle displacement, is a function of time; while at any instant in time the displacement at a point is a function of the position of the point.

Waveform

Instantaneous amplitude as a function of time.

Wavelength

The wavelength of a pure tone is the distance between analogous points on two successive waves. Wavelength (λ), frequency (f), and the speed of sound (c) are related by the expression $c = f\lambda$. In most practical situations, the sound of interest is more aptly described as noise (random noise) than as a finite composite of pure tones.

Weighting

Prescribed frequency response provided in a sound-level meter.